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76-INCH DIAMETER CENTRIFUGE FACILITY

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PREFACE

The purpose of this Technical Note is to serve as an operating guide for the 76-inch diameter centrifuge facility. The operating procedures recommended in this manual evolved during the initial six-month operating period during which propellant burning rate experiments were conducted. The recommended procedures have been found to be both safe and efficient.

The writers wish to express their appreciation to Mr. Norman Walker, Civilian Supervisor of the U. S. Naval Postgraduate School Machine Facility, and his staff, for the construction of the centrifuge; and Mr. Edward J. Smith for his assistance in the installation and operation of the centrifuge equipment.

76-inch Diameter Centrifuge Facility

TABLE OF CONTENTS

	Page
Equipment	1
Operation	12
Maintenance	26
References	34
Appendices	
A. Determination of Acceleration Field	35
B. Calculation of Pressure Change in Combustion Bomb due to Centrifugal Force	39
C. List of Drawings	45
D. Stress Analysis	47

LIST OF PLATES

PLATE PAGE

1. 76-inch Diameter Centrifuge Facility, U. S. Naval Postgraduate School, Monterey, California.......... 78

2. Centrifuge Control Station............................79

3. Centrifuge Rotor......................................80

4. Combustion Bomb-Surge Tank System (Rotor System)....81

5. Strand Holders Used With Electric Timers in Burning Rate Experiments.....................................82

6. Strand Holders Used With Pressure Instrumentation in Burning Rate Experiments..........................83

7. Strand Holder and Bomb Assembly......................84

8. Centrifuge Lower Shaft Detail........................85

9. Centrifuge Base Structure............................86

10. Pressure Circuit.....................................87

11. 12 v. d.c. Electrical System.........................88

12. 115 v. a.c. Electrical Circuit.......................89

13. Timing, Ignition, and Continuity-Test Circuits......90

14. Visicorder Timing Circuit............................91

15. Vibration Pickup Circuit.............................92

16. 6 v. d.c. Electrical System..........................93

17. Nitrogen Manifold....................................94

18. Charging Station.....................................95

19. Centrifuge Drive Unit................................96

20. Centrifuge Hydraulic Throttle Control 97

21. Centrifuge Brake System 98

22. Visicorder Signal Conditioning Panel 99

23. Visicorder Signal Conditioning - R_1 versus V_1 ... 100

24. Visicorder Signal Conditioning - R_1 versus R_2 ... 101

25. Hoist and Sling Attached for Removal/Installation of Rotor.. 102

26. Rotor Balancing Stand............................ 103

LIST OF FIGURES

FIGURE		PAGE
A1	Radial Acceleration G at 35.6-inch Radius.........	36a
A2	Radial Acceleration G_r at 35.6-inch Radius........	37
A3	Angular Displacement of the Resultant Acceleration Vector G with Changing Radial Acceleration G_r	38
B1	Pressure Change at 35-inch Radius - Two Surge Tanks	42
B2	Pressure Change at 35-inch Radius - One Surge Tank	44
D1	Peak Loading Condition on Centrifuge Structure ...	48
D2	Peak Loading Condition on Centrifuge Shaft........	51

LIST OF FIGURES

FIGURE PAGE

A1 Radial Acceleration G at 35.6-inch Radius........ 36a

A2 Radial Acceleration G_r at 35.6-inch Radius....... 37

A3 Angular Displacement of the Resultant Acceleration Vector G with Changing Radial Acceleration G_r.................................. 38

B1 Pressure Change at 35-inch Radius - Two Surge Tanks.................................. 42

B2 Pressure Change at 35-inch Radius - One Surge Tank.................................. 44

D1 Peak Loading Condition on Centrifuge Structure.................................. 48

D2 Peak Loading Condition on Centrifuge Shaft....... 51

EQUIPMENT

The 76-inch diameter centrifuge show in Plate 1 was designed and constructed at the U. S. Naval Postgraduate School, Monterey, California. The centrifuge is installed in an experimental test cell at the School's Rocket Laboratory. The configuration shown was designed to study the burning rate of solid propellants in acceleration fields up to 2000G. The 1665 cu. in. bomb-surge tank system on the rotor was designed for operating pressures up to 3000 psig. The centrifuge could be used for other purposes by fitting a suitable rotor to the vertical shaft. The centrifuge capacity is 36,000 g-pounds at a maximum speed of 1450 RPM.

A 12 channel slip-ring assembly is provided for instrumentation on the rotor. Pressure in the combustion bomb is sensed by a pressure transducer.

Propellant burning rate may be determined by two methods. Three timers are used to record elapsed burning time over known distances, and the total burning time of the sample may be determined from the pressure-time history recorded on a Visicorder chart.

The centrifuge is operated from the remote control station shown in Plate 2.

Centrifuge

The centrifuge base structure is of welded and bolted construction. The base was constructed from standard 12 inch

channel and wide-flange beam stock. The struts supporting the upper bearing housing are standard steel pipe.

The centrifuge shaft rotates in two bearings. The lower bearing is an SKF self-aligning double-row ball bearing, and the upper bearing is an SKF spherical roller bearing (self-aligning). Loads from the lower bearing are transmitted through the base structure to a thrust pad in the floor. Loads from the upper bearing are transmitted to the walls of the test cell through the tubular struts.

The structural parts of the rotor are made of aluminum, and are bolted together with aircraft quality steel bolts. The rotor arms are constructed from 3/4 inch plate. To these main structural members are secured the combustion bomb, surge tanks and connecting tubes, fittings and valves, pressure transducer, and counterweight.

The rotor assembly is secured to the shaft by a pivot pin. Thus the rotor is dynamically self-balancing. Static balancing is accomplished by setting the rotor on a balancing stand and adding or removing counterweight plates.

Aerodynamic drag has been reduced to an acceptable level by fairings on the rotor arms.

<u>Combustion Bomb-Surge Tank System</u> <u>(Rotor System)</u>

The bomb-surge tank system is wholly contained in the rotor assembly which is shown in Plate 3. It consists of the combustion bomb, two surge tanks, and connecting tubes and valves. Hereafter, this system will be referred to as the Rotor System.

The combustion bomb was made from three pieces of 321 stainless steel. A hemispherical cap and a reducer section were welded to a cylindrical center section. The reducer section has a 1 7/8-inch I.D. port to permit insertion of the strand holder. The strand holder also serves as a plug and is locked in place with an aluminum collar. The bomb has an inside diameter of 4 inches and a capacity of 115 cu. in. It was designed for a working pressure of 3000 psi and has been hydrostatically tested to 4500 psi.

The two surge tanks are Navy SCUBA tanks with a capacity of 725 cu. in. each. They are made of 6061-T6 aluminum and were designed for a working pressure of 3000 psig. The tanks are lined with Dow Corning Q 92-009 silicone rubber sealant for protection against corrosion.

The tubing used in the system is seamless, annealed, 321 stainless steel. The bomb and surge tanks are connected by 0.375-in. O.D. x 0.258-in I.D. tubing. A ball-valve is located between the bomb and the tanks.

The system may be pressurized through a quick-disconnect fitting located between the ball-valve and the surge tanks.

The system may be de-pressurized by a hand operated discharge valve or a solenoid valve actuated by a switch on the control console. Both valves are connected to the bomb by a 0.250 in. O.D. x 0.180 in I.D. tube to a tee in the discharge line.

A diagram of the Rotor System is shown in Plate 4.

Strand Holders

The strand holders shown in Plates 5, 6, and 7 are all of the same general type. The basic components are a machined aluminum plug, convas phenolic slab and strand support, insulation sheet, and gland seal. The aluminum plug is a slip fit in the throat of the bomb and the flared edge at the top of the plug is a metal-to-metal fit relative to the bomb throat seat. Thus the strand holder is supported in high radial acceleration fields, and extrusion of the O-ring seal is avoided.

The slab and strand support, both of canvas phenolic, provide structural support for the propellant sample. The slab is fastened to the plug with two #10-32 FH screws, and the strand support is bonded to the slab with APCO 210 Resin and 180 Hardener. #6-32 brass machine screws are used for timing and ignition circuit terminals. The terminals are connected to an Amphenol plug by #20 enameled copper magnet-wire. The copper wires are bonded to the back of the slab with APCO epoxy to protect them from contact with the combustion products.

Strand-holders for use with the electric timers have five pairs of terminals on the slab; four for the timing circuits and one for the ignition circuit. One terminal of each pair is a common ground to the aluminum plug. The remaining five leads are connected to the Amphenol plug through a single 1/8 in. dia. gland in the aluminum plug. The gland was reamed on the inside with a 10^{o} tapered reamer so that the inside of the hole looked like a long thin funnel. With the electrical

leads installed, the remaining space was filled with APCO epoxy using a hypodermic syringe and needle. The epoxy provides an effective gland seal for internal pressures up to 1000 psig. At 1500 psig some leakage occurs.

A commercial four-wire transducer gland seal (Conax MTG-20-4) is used with one of the strand holders with five pairs of terminals. The pins of the Amphenol plug for the last timing wire and the ignition wire are connected by a jumper, and the terminals for the last timing wire and the ignition wire are connected by a 150 Ω , ½ watt resistor. When the strand ignition switch is actuated a small current flows in the last timing wire, but not enough to cause a significant increase in wire temperature. A current reversal in the holding coil of timer relay No. 4 is prevented by a diode in the holding coil circuit (see Plate 13). When the ignition wire burns through and the ignition switch is released, sufficient current flows through the 150 Ω resistor (with the last continuity wire intact) to actuate relay #4.

The essential difference between the strand holders designed for use with the electric timers and those designed for use only with the pressure instrumentation is that the latter have only one pair of terminals. Also, the Conax gland seals are used exclusively in the two-wire strand holders.

The canvas-phenolic slab is protected from direct contact with the exhaust flame by a 3/32 in. thick phenolic plastic insulation sheet (FSN 9330 282 5641). The insulation sheet covers

the entire slab and is held in place by the ignition wire terminal nuts and a #3-48 screw near the aluminum plug.

For acceleration normal to and into the burning surface, propellant samples are placed on the strand holder with their bases resting on the strand support as shown in Plates 6 and 7. Masking tape wrapped around the sample and strand holder holds the sample securely in place at all acceleration levels up to 2000G.

Instrumentation and Electrical

RPM instrumentation consists of a SPACO type PA-1 Magnetic Pickup and a Berkeley Model 5545 EPUT Meter. The timing disc has 30 teeth and is mounted behind the shear coupling. See Plate 8. With a counting period of one second, RPM/2 is read on the EPUT Meter.

A Lebow Model 6109-12 instrumentation slip ring assembly is mounted on the centrifuge shaft above the brake disc. The assembly is rated for 2000 RPM maximum and has coin-silver rings and silver-graphite brushes for low noise. Electrical leads from the ring terminals pass through two holes in the hollow shaft and connect to the underside of a circular terminal block in the top of the shaft. The slip ring assembly is shown in Plate 9.

Pressure in the combustion bomb is sensed by a Daystrom-Wiancko Type P2-3086 or P2-1251 variable-reluctance pressure transducer. The transducer is mounted at the centrifuge axis of rotation in order to minimize the effects of acceleration

on the instrument. 28 v. d.c. excitation is provided by a Hewlett-Packard Model 721A power supply. The zero to 5 v. d.c. output is read on a Weston Model 911 D.C. Voltmeter having an impedance of 20KΩ/volt.

Pressure change in the bomb is recorded on a Honeywell Model 1508 Visicorder. A bucking voltage is applied to the transducer output with a 6 volt dry cell battery and a potentiometer, and the resultant small signal then goes to a Kintel Model 112A single-ended D. C. Amplifier. Amplifier output is fed to an M400-120 galvanometer in the Visicorder. Variable series and shunt resistors in the galvanometer circuit permit adjustment of chart span and galvanometer damping. Zero adjustment is provided by changing the bucking voltage potentiometer setting. A schematic drawing of the circuit is shown in Plate 10.

Average burning rate may be determined by recording elapsed time during burning over a known distance. The primary means utilizes the pressure-time history recorded on the Visicorder chart. Burning is assumed to commence when the pressure begins to rise, and end when the pressure stops rising. Dividing the initial sample length by the total burning time gives the average burning rate.

The secondary means utilizes timing wires and electric timers. Three Standard Electric Time Co. Model S-1 Timers are used for recording elapsed time. 115 v. a.c. power to the timer motor clutch coils is controlled by four Magnecraft W133MPCX-3 mercury-wetted-contact SPDT relays.

Four timing wires are equispaced in the propellant sample. The timing wires are 0.008 inch diameter Pyro-fuse, which is composed of an aluminum core sheathed in palladium. At 600°F., the metals alloy rapidly and exothermically, thus breaking the 6 v. timer relay holding-coil circuits.

The propellant sample is ignited by a #32 (0.008-in. dia.) nichrome resistance wire. Power is supplied from the 12 v. d.c. bus.

Continuity of the ignition and timing circuits can be checked by means of a rotory switch which applied 6 v. d.c. power to five continuity test lamps. Schematic drawings of the ignition, timing, and continuity test circuits are shown in Plates 11, 12, 13 and 14.

Vibration of the centrifuge structure may be monitored from the control station by reading on a Honeywell Model 1508 Visicorder the output from a vibration pickup. The pickup is a Statham Model A5A-15-350 accelerometer (± 15 g) which is mounted on the centrifuge structure near the upper bearing (Plate 3). 10 v. d.c. excitation is provided by the 28 v. d.c. output from a Hewlett-Packard Model 721A power supply in series with a dropping resistor. The ± 4 mv. output from the accelerometer goes directly to an M200-120 galvanometer in the Visicorder. A schematic drawing of the circuit is shown in Plate 15. A microphone is located just below the lower bearing at the base of the centrifuge. Audio noise thus picked up passes through an amplifier to earphones worn by the operator.

The engine electrical system is a standard 6 v. automobile system. A schematic drawing is shown in Plate 16. A four position ignition switch (ACC, OFF, ON, START) is located at the control console. See Plate 2. The ACC position provides power to the 6 v. dc bus when the engine is not running. Stewart-Warner oil pressure and water temperature gages and an ammeter are installed in the control console. The oil pressure and water temperature gages are electrically operated and receive power from the 12 v. bus. A bulb type gage secured to the side of the Powerglide transmission indicates transmission fluid temperature. Instrumentation noise from the engine electrical system is reduced to an acceptable level by a 0.2 micro-farad capactor connected to the generator armature terminal and a radio noise suppresor in series with the coil high-tension lead.

Nitrogen Charging System

The nitrogen charging system consists of a four-bottle manifold, regulator, gage panel, and flexible charging hose. The manifold and gage panel are shown in Plates 17 and 18. The manifold is a Victor "Simplex" discharging type with a valve for each bottle and a master shut-off valve at the manifold outlet. The pressure regulator is a Victor GD 31 Gas-o-dome regulator, rated for 3600 psi inlet and outlet pressure.

On the gage panel are mounted three Marsh Type 220-3S pressure gages. The gages are the bourdon-tube type with a 6-inch dial. Pressure ranges are 0-1000 psig, 0-3000 psig, and 0-5000 psig. Suitable valves permit the selection of the appropriate gage for use in pressurizing the surge tanks and combustion bomb. A flexible charging hose is used to connect the

gage manifold to the rotor system. On the end of the hose is a Bruning "Magnum" 251 SS-456-E22 coupler body. The poppet valve has been removed from the coupler body to permit connection of the charging hose while the surge tanks are pressurized.

Connecting tubing used in the system is the same as that used in the Rotor System.

Drive System

The centrifuge is driven by a 1954 Chevrolet engine with a 1950 Powerglide transmission. Power is transmitted to the centrifuge via an automobile drive shaft and a Boston VR158 Sprial Miter-Gear Box. A 285 ft-lb shear coupling is fitted on the miter gear input shaft. The four shear pins are made from 1/8" diameter brazing rod. The coupling is designed so that it will remain intact after the pins have sheared. A double-strand roller-chain flex-coupling transmits the torque from the miter gear vertical output shaft to the centrifuge shaft. The coupling is made from two Boston 6.37 P.D. steel sprockets and No. 40-2 standard roller chain.

Controls

The engine throttle setting is controlled by an Adel ISOdraulic Remote Control System. The installation is shown in Plates 2 and 19. A diagram of the system is shown in Plate 20.

The centrifuge is equipped with hydraulically actuated disc type brake. The brake is actuated by a foot pedal, and

a hand operated cam is used to lock the brake on the ON position. A hand operated relief valve is installed in the brake line. In the OPEN position, the valve vents the fluid in the line back to the reservoir thus avoiding a hydraulic lock in the brake system due to thermal expansion. The brake components are shown in Plates 2, 8, and 9, and a diagram of the system is shown in Plate 21.

A deadman switch is incorporated in the engine ignition circuit as a safety percaution.

OPERATION

General

The centrifuge is operated from the control station in Room 108. It can be operated by one person, but it has been found that propellant burning rate experiments can be more closely controlled with two people. One operator controls the throttle and brake while monitoring the RPM counter, engine instruments, and audio noise. The other sets up the burning rate instrumentation and monitors the accelerometer output on the Visicorder.

Instrumentation should be turned on to permit stabilization at normal operating temperatures before actual experiments are conducted. This is particularly true for the Kintel Model 112A amplifier which requires about a 30 minute warm-up time. In addition, the following items should be checked before operating the centrifuge.

1. Transmission in Park or Neutral
2. Proper oil level in the miter gear box and engine crankcase
3. Transmission fluid between ADD and FULL
4. Sufficient water in the radiator and storage batteries
5. Sufficient fuel
6. Rotor balanced, free of hoses, tools and other loose items, and securely attached to the shaft
7. No loose gear adrift in the test cell. Hoses, slings, rags, etc., must be securely stowed to prevent their being drawn into the path of the rotor.

8. Area around engine clear

9. Exhaust pipe connected

10. Warning signs in place

Drive System

Starting the engine. The engine is normally operated from the control station. The 12 v. d.c. master switch must be turned ON in order to operate the water-temperature and oil-pressure gages. The deadman switch, which is located on the floor to the left of the brake pedal, must be CLOSED in order to complete the ignition circuit. The switch can be closed by stepping on the foot pedal or by turning ON the red-guarded over-ride switch. With the red-guard DOWN, the over-ride switch is OFF.

The ignition switch is located at the lower right-hand side of the console. It has the following four positions from left to right: ACC, OFF, ON, START.

The engine can also be cranked by closing a switch on the starting relay. The relay is mounted on the engine base below the starting motor.

> CAUTION: with the engine running and the transmission fluid cold, there is sufficient torque transmitted to the transmission output shaft to turn the centrifuge rotor even with the transmission selector in NEUTRAL. With the selector in PARK, no torque is transmitted to the transmission output shaft.

Engaging the transmission. The transmission is controlled by a single selector-lever located on the left hand side below

the hydraulic throttle control slave unit. The five positions (rotating the selector shaft counter-clockwise are PARK, NEUTRAL, DRIVE, LOW, and REVERSE. With the selector in PARK, the transmission output shaft is locked to the transmission case.

NOTE: The rotor should be checked for freedom of rotation before moving the selector to DRIVE.

The selector should be moved from NEUTRAL to DRIVE or LOW with the engine speed not above idle. For speeds up to about 1200 RPM, the selector should be placed in DRIVE. For speeds above 1200 RPM, the selector should be placed in LOW. LOW may be used at speeds below 1200 RPM if desired, but operation in LOW tends to cause the transmission to run hotter than in DRIVE. When LOW is used, the transmission fluid temperature gage should be checked frequently to prevent over heating. The maximum permissible temperature is 240^{o}F.

<u>Operation</u>. With the transmission engaged, all subsequent control is exercised from the control station. With the centrifuge running at idle, the brake-line relief valve (see Plate 21) should be left OPEN momentarily to ensure running clearance between the brake pucks, and then CLOSED. The valve is OPEN with the handle in line with the connecting tube, and CLOSED with the handle ninety degrees to the connecting tube.

The brake is applied by stepping on the brake pedal. One or two pumps may be required to obtain maximum braking. With the brake applied, pulling the hand lever toward the operator will keep the brake applied.

NOTE: the brake is provided for stopping the centrifuge, and NOT for speed control. As in an automobile, "riding" the brake results in overheating and possible damage to it and adjacent components. After each brake application, the brake line relief valve should be opened momentarily to regain running clearance.

With the brake-line relief valve closed, the deadman switch engaged (operator's foot on the switch pedal and red guard down), and the EPUT Meter and engine instruments operating, the centrifuge may be accelerated to the desired speed.

NOTE: the EPUT Meter reads RPM/2. e.g., with 500 showing on the EPUT Meter, the RPM is 1000. Operating instructions are contained in the instrumentation section.

While the centrifuge is running the operator should monitor the EPUT Meter, engine instruments, and be alert for the following:

1. Lack of response to throttle position
2. Unusual noise

Either of the above two symptoms is cause for immediate shut down by closing the throttle, turning off the ignition, and applying the brake.

The first symptom could be caused by an engine failure; most likely fuel starvations or an opening in the ignition circuit. In this event, of course, no action by the operator would be necessary. It could also be caused by failure of the

pins in the shear coupling. In this case, stopping both the engine and centrifuge as soon as possible will minimize wear in the coupling.

A gradual reduction in speed with a fixed throttle setting may be caused by a dragging brake. Opening the brake line relief valve for a few seconds will regain running clearance.

The second symptom could be due to a failure in the miter gear, flex coupling, bearings, or rotor assembly.

It should be noted that at high speed, audio noise from the engine and the forced vortex in the test cell due to the rotor motion are more audible to the operator not wearing the earphones. This is particularly true at speeds above 1000 RPM.

Shut down. The centrifuge is normally stopped by closing the throttle and allowing the machine to slow down due to air resistance. The ignition switch may be turned OFF before or after the transmission is taken out of gear. In either case, the transmission selector should be placed in NEUTRAL (or PARK if the rotor has stopped) before going into the centrifuge area.

To secure the centrifuge the following should be accomplished:

1. Throttle CLOSED
2. Transmission selector NEUTRAL or PARK
3. Ignition OFF
4. 12 v. master switch OFF
5. Brake line relief valve OPEN
6. EPUT Meter and audio noise amplifier OFF

Nitrogen System

The nitrogen system includes the Charging System and the Rotor System. The Charging System consists of a Victor Simplex four-bottle manifold which is fitted with a Victor GD-31 regulator, a pressure-gage panel, connecting valves and tubing, and a flexible charging hose. The Rotor System is located on the centrifuge rotor. It consists of the combustion bomb, two surge tanks, and connecting valves and tubes.

Manifold and regulator. The manifold has a shut-off valve for each of the four bottles and a master valve at the manifold outlet (Plate 17). To change a bottle, both the shut-off valve and the valve on the bottle must be CLOSED. The handle on the pigtail (tube connecting the bottle to the manifold) must be held firmly while tightening or loosening the gland nut.

Pressure is supplied to the regulator by pressurizing the manifold and opening the main valve at the manifold outlet. Supply pressure is indicated by the lower gage on the regulator.

> CAUTION: The position of the valves at the charging station should be checked before opening the manifold main valve.

The regulator is controlled by two valves, a load valve and a bleed valve. Opening the load valve supplies gas pressure to the dome, thus increasing delivery pressure. To decrease the delivery pressure, the load valve must be readjusted and the excess gas in the dome vented through the bleed valve. Opening the bleed valve vents the dome only. Down stream gas

must be vented through the valves at the charging station.

Charging station. The pressurization of the Rotor System is controlled from the Charging Station. A fill valve admits the nitrogen gas to the gage manifold and charging hose, and a bleed valve vents the gas to the atmosphere. The 0-5000 psig gage indicates at all times the gas pressure upstream from the fill valve (regulator delivery pressure). The 0-1000 psig and 0-3000 psig gages indicate the pressure in the gage manifold and charging hose only when their shut-off (s.o.) valves are open. An additional vent valve is provided for the protection of the 0-1000 psig gage. Normally, the valves are left in the following positions:

Fill valve .... CLOSED
Bleed valve .... OPEN
0-3000 psig gage s.o..... OPEN
0-1000 psig gage .... CLOSED
vent .... OPEN

Rotor system. The rotor System is charged through a quick-disconnect fitting. Discharging can be accomplished two ways; through a solenoid valve controlled by a switch on the control console, and by a hand-operated discharge valve. The solenoid valve is provided for discharging high pressure gas (above 2000 psig) by remote control. A detachable discharge line is connected to the outlet of the hand operated discharge valve for discharging. The line carries the gas outside the test cell.

The bomb can be isolated from the surge tanks by closing the ball-valve, thus permitting nitrogen to be retained in the surge tanks while the bomb is depressurized. With the ball valve closed, only the surge tanks can be charged. The surge tanks can be discharged only through the bomb.

Charging the rotor system. Prior to the installation of the strand holder in the bomb the Nitrogen System is normally set up as follows:

Charging Station

Fill valve ....CLOSED

Bleed valve ....OPEN

Gage shut-off valves ....as desired

Manifold and Regulator

Manifold ....Pressurized

Main valve ....OPEN

Regulator supply pressure ....as desired

Rotor System

Ball-valve ....CLOSED (clock-wise)

Solenoid valve ....CLOSED

Discharge valve ....OPEN (line attached)

After the strand holder and retaining collar are installed, the discharge valve is CLOSED and the line removed. The charging hose is then connected.

At the charging station the bleed valve is CLOSED and the fill valve OPENED. Charging rate and final pressure are controlled by the fill valve. When the pressure in the charging

hose exceeds the pressure in the surge tanks by 8 psi, the tanks will begin to fill. The ball-valve is then opened slightly to permit fresh nitrogen to fill the bomb. This can be accomplished by regulating the fill valve such that the charging hose pressure increases slowly while the pressure in the bomb and surge tanks is equalizing. After the pressure in the bomb and surge tanks has equalized, the ball-valve is OPENED FULL(counter clock-wise).

When the desired pressure is attained in the bomb and surge tanks, the fill valve is CLOSED and the bleed valve OPENED thus discharging the gage manifold and hose. The charging hose is then uncoupled from the rotor.

NOTE: With a very low flow rate, the pressure read on the gages at the charging station is 8 psi greater than the pressure in the bomb and surge tanks.

Discharging the rotor system. To discharge the system, the discharge line is hooked up to the discharge valve, and the valve opened slowly. Opening the valve wide with over 500 psi in the system may cause the flexible discharge line to rupture. If the bomb only is to be depressurized, the ball-valve is closed. Discharging for a few seconds before closing the ball-valve value will purge the line connecting the bomb to the surge tanks.

After the bomb has been depressurized the strand holder can be removed.

Securing the nitrogen system. The nitrogen system is normally secured in the following manner:

Rotor System

Ball valve .... CLOSED

Solenoid valve .... CLOSED

Discharge valve .... OPEN (line attached)

Charging Station

Fill valve .... CLOSED

Bleed valve .... OPEN

Gage s.o. valves .... as desired

Manifold

Main valve .... CLOSED

Shut-off valves .... CLOSED

Instrumentation

Instrumentation is separated into two categories; one associated with the operation of the centrifuge and the other associated with the determination of propellant burning rate.

Centrifuge. Centrifuge instrumentation consists of the engine instruments, tachometer, audio noise system, and vibration pickup.

The engine oil-pressure and water-temperature gages are powered from the 12 v. d.c. bus. Hence the 12 v. d.c. master switch must be turned ON to operate these instruments. The transmission temperature gage is the bulb type and requires no external power.

The tachometer consists of a magnetic pickup and a Berkeley EPUT meter. With a counting time of one second, RPM/2 is read on the counter. To operate the system, the position of the switches on the meter should be as follows:

Unit time 1

Auto - Manual .... AUTO (up)

Ext. start - Int. start ...... INT START (down)

Operate - Test .... OPERATE (up)

Counter - EPUT .... EPUT

Count .... OFF

Input .... TACHOMETER

Power .... ON

To secure the EPUT meter, the power switch is turned OFF. All other switches may remain in their operating position.

The audio noise system is operated by simply turning on the amplifier and adjusting the gain as necessary.

The vibration pickup is an accelerometer attached to the upper bearing support. The required 10 v. d.c. power is supplied to the accelerometer by turning on the Hewlett-Packard Model 721A power supply and adjusting the output to 28 v. d.c. A variable dropping resistor on the Visicorder signal conditioning panel is adjusted to provide the required 10 v. d.c. drop across the accelerometer.

Output from the accelerometer is read on the Honeywell Model 1508 Visicorder. Complete operating and maintenance instructions for the Visicorder are contained in Reference 1.

Burning rate. Instrumentation is provided for determining burning rate by two methods; recording the pressure-time history throughout the burning of the sample, and electric timers to record, the elapsed time of burning between timing wires spaced

at known intervals along the sample axis. The two independent systems are coupled by providing a separate galvanometer in the Visicorder for each timing wire, such that the continuity of each wire is indicated by a separate trace on the Visicorder chart.

To operate the pressure trace system, the following components must be turned on and allowed sufficient warm-up time (about 20 minutes):

Hewlett-Packard Model 721A power supply ....28 v.

Kintel Model 112A amplifier ....Gain zero

Honeywell Model 1508 Visicorder ....Main power ON (See Ref. 1 for Oper Inst.)

The signal conditioning circuit mounted on the rear of the Visicorder (Plate 22) permits the output from the pressure transducer to be nulled before the signal is fed to the amplifier. It also permits adjustment of the chart span to the anticipated pressure rise.

The chart span is adjusted as follows:

1. The expected transducer voltage output rise (mv) is calculated from the expected pressure rise and the transducer pressure and output voltage ranges.
2. The corresponding value of R_1 for a six-inch galvanometer deflection is determined from Plate 23.
3. The required value of R_2 for proper galvanometer

damping is then obtained from Plate 24.

CAUTION: When checking R_2 the galvanometer must be disconnected to prevent damage to the galvanometer and to obtain a correct resistance reading. When checking both R_1 and R_2 the amplifier output cable must be disconnected in order to obtain a correct reading.

With the desired pressure in the bomb, the transducer output signal is nulled as follows:

1. Amplifier input switch - OFF
2. Dry cell battery behind panel connected to circuit using alligator clips
3. Move selector switch to BATTERY (up) and adjust potentiometer so that voltmeter on control console reads near zero
4. Move selector switch down to XDCER, press Null Test button below the microammeter and adjust potentiometer to center microammeter needle

To put the signal into the Visicorder, the amplifier gain is set at 20 and the input switch is turned ON. Trace position is adjusted with the same potentiometer used for nulling the transducer output. A pressure increase causes the trace to move to the right, while a pressure decrease causes the trace to move to the left.

CAUTION: The galvanometers are easily damaged by too high a voltage. The input switch should not be left on any longer than necessary.

The equipment is secured as follows:

Input switch .... OFF

Amplifier .... OFF, Gain zero

Visicorder .... OFF (Refer to Reference 1 for recommended procedure.)

Power supply .... OFF

Null bettery .... DISCONNECTED

To operate the timing wire system the following must be turned on:

Visicorder

6 v. d.c. bus (ACC or ON)

115 v. master ....ON

Clock master ....ON

Continuity of the ignition and four timing circuits is checked by rotating the continuity switch to each individual circuit. A greenlight indicates continuity. The continuity switch should be returned to OFF after completing the check. Continuity of the timing circuits is also indicated by the appearance of four galvanometer spots at the right-hand edge of the Visicorder chart.

The timers are individually set to zero by pressing down on the lever at the upper left hand corner of the timer.

Engine

The engine is a 1954 Chevrolet powerglide engine with hydraulic valve lifters. Preventive maintenance includes maintaining the proper water levels in the battery and radiator, maintaining the proper oil level in the crankcase, periodic oiling of the generator (two oil cups), and periodic lubrication of the distributor shaft via the grease cup on the shaft housing.

An automobile repair manual should be consulted for engine specifications and additional maintenance instructions.

NOTE: The carburetor float level has been adjusted to give the highest possible fuel level in the bowl. This has been found to produce the smoothest idle.

Transmission

The transmission is a 1950 Chevrolet Powerglide. A 1955 Pontiac open drive line adaptor is fitted on the output end of the transmission. A temperature gage is mounted on the left-hand side of the transmission, and the bulb is immersed in the transmission fluid reservoir. Fluid temperatures below 240°F are normal. Fluid level should be measured with the engine running and the transmission warm.

Universal Joints

A standard Chevrolet universal joint is located at each end of the 1954 Corvette drive shaft. The joints are packed with grease, ball and roller bearing (MIL-G-18709(NAVY). Grease can be added through the zerk fittings on each joint.

Shear Coupling

The shear coupling has four shear pins made of 1/8 inch diameter brazing rod. In the event the pins shear, the coupling should be checked for freedom of relative rotation between the coupler halves. Any sign of binding is cause for disassembly, cleaning, and repacking with grease.

Miter Gear

The miter gear is a standard Boston Model VR158. It is lubricated with SAE 20W detergent type automobile engine oil. The proper oil level is at the bottom of the filler plug (located at the rear of the unit) with the oil cool and the unit not operating. Oil changes should be made after the first 80 hours of operation, the next 250 hours, and subsequently every 1000 hours. The oil should be completely drained, preferably while warm, and then refilled with fresh SAE 20w detergent motor oil. The capacity of the unit is one quart, eight ounces.

Flexible Coupling

The flexible coupling is packed with grease. The grease is retained by a rubberized cotton boot which should be removed for inspection and repacking approximately every five years.

Shaft Bearings

The bearings are lubricated with Grease, Ball and Roller Bearing (MIL-G-18709(NAVY)). According to Reference 2, this is adequate for sustained operation up to 1000 RPM and intermittant operation up to 1500 RPM. Zerk fittings are provided for adding grease, but it should be born in mind that the bearings

require very little lubrication. If too much grease is injected it will be "pumped" as the bearings roll around the race. This can result in overhearing sufficient to damage the bearings.

Lower bearing - removal

1. Remove Rotor Assembly from centrifuge shaft as described in the next section. Set Rotor Assembly on floor and remove hoist from Rotor.
2. Clean grease and dirt from vicinity of upper bearing, shaft, lower bearing, flex coupling and gear box.
3. Unlace flex coupling grease retainer and remove retainer.
4. Remove spring clip from top of connecting link in roller chain and remove chain.
5. Clean grease from between sprockets. Loosen set screw in lower sprocket thus permitting lower sprocket to slide up and down on gear box output shaft.
6. Remove four bolts holding upper sprocket to centrifuge shaft. Two clearance holes are provided in the lower sprocket to facilitate removal of the bolts.
7. Remove the three bolts holding the brake mounting bracket to the lower bearing housing.
8. Remove all socket head screws holding the upper bearing housing to the upper bearing support.
9. Secure the chain hoist to the shaft. Use extreme care not to damage the pin bearing surfaces in the shaft.
10. Raise the shaft approximately one-half inch and move the brake caliper away from the brake disc.

11. Raise the shaft clear of the lower bearing.

    CAUTION: Vertical travel is limited by the slip ring assembly and electrical wiring. Neither should be permitted to contact the underside of the upper bearing support.

12. Remove the remaining bolts in the lower bearing housing and remove the housing, being careful to keep track of shims.

Rotor Assembly

Removal. The rotor assembly is removed from the centrifuge shaft as follows:

1. Disconnect Bendix plug from transducer.
2. Disconnect remaining electrical circuits at the top-shaft terminal block.
3. Secure the overhead chain hoist and sling to the rotor as shown in Plate 25.
4. Raise the hoist just enough to remove the rotor load from the pivot pin.
5. Remove one nut and washer from the pivot pin and slide the pin out BY HAND.

    NOTE: If the pin will not slide out easily, the hoist tension is not correct.

6. Raise the rotor clear of the shaft.

    CAUTION: Care must be used in raising the rotor to prevent damage to the electrical wires. The rotor must be steadied as it clears the shaft to prevent

sideways motion and subsequent damage to the binding posts in the terminal block.

Balancing. The rotor is balanced statically using the Balancing Stand shown in Plate 26. For longitudinal balance, the pivot pin is centered on the stand and permitted to roll. Care should be taken to ensure that the surfaces of the pin and stand are clean, and the stand level. Plates are added to or removed from the counterweight in order to achieve static balance. Lateral balancing is accomplished by using the Balancing Pin. The lateral pin can be centered using adjustable parallels. Location of the lateral pin is critical and should be checked after it is resting on the balancing stand. The surge tanks may be shifted in order to achieve lateral balance.

To facilitate rebalancing, the removable counterweight plates have been numbered in the upper left hand corner (looking inboard from the counterweight to the center of rotation).

With both surge tanks connected, transducer c'wt.* removed, and the rotor system pressurized, the proper weight combinations for static balance are as follows:

Pressure, psig	Plate Nos.
0	1, 2, 3, 4, 11.
500	1, 2, 3, 4, 11, 500.
1000	1, 2, 3, 4, 11, 1000.
1500	1, 2, 3, 4, 11, 1000, 500.

* The transducer c'wt. is a steel block which replaces the potentiometric transducer shown on Dwg. 1162.

With the surge tank on the bomb end disconnected, transducer c'wt. installed, and the system pressurized, the proper weight combinations are as follows:

Pressure, psig	Plate Nos.
0	1, 2, 3, 4, 11.
500	1, 2, 3, 6, 7, 8, 9, 500.
1000	1, 2, 3, 6, 7, 8, 9, 10, 11.
1500	1, 2, 3, 6, 7, 8, 9.

Installation. The rotor is installed on the shaft using the overhead chain hoist. The arrow on the top shaft terminal block should point toward the bomb.

Care should be taken to line up the holes in the rotor arms with those in the shaft. Prior to inserting the pivot pin, its surface and the bearing surfaces in the rotor and shaft should be cleaned and lightly coated with grease.

NOTE: Use no tools to insert the pivot pin.

The pin will slide in by hand if the rotor and shaft are properly aligned.

The nuts on the end of the pin should be tightened just enough to eliminate end play in the pin.

Hydraulic Throttle Control

The engine throttle is operated by an Abel ISOdraulic Remote Control System. The system consists of two 180° units - one master and one slave, and a compensator. The compensator performs two functions: it applys a pre-load (pressure) to the system and compensates for relative volume changes in the system.

The compensator is also used for filling and bleeding the system.

Only clean, Shell Diala Oil AX (a transformer oil) should used in the system. Complete instructions for bleeding, filling and pressurizing, and synchronizing the system, as well as maintenance and trouble shooting are contained in Reference 3.

<u>Brake</u>

The fluid level in the brake reservoir should be checked periodically. The fluid is at the proper level when the reservoir is approximately one-half full. Only the type fluid indicated on the reservoir should be used.

Two people are required to bleed the brake. The system can be bled as follows:

1. Check reservoir for proper fluid level.
2. Close the relief valve (see Plate 21).
3. Fasten one end of a short length of flexible hose to the bleed valve (Plate 8) and put the other end at the bottom of a container.
4. While one person pumps the brake pedal, the other opens the bleed valve ¼ to ½ turn.
5. When the fluid flows without entrained air bubbles, the bleed valve is closed.
6. Test brake. The pedal should feel firm when the brake is applied. Sponginess indicates that air is still in the system and the brake should be bled again.

Slip Ring Assembly

The Lebow Model 6109-12 slip ring assembly is a self-contained unit. It is rated for a maximum speed of 2000 RPM. The shielded ball bearings are factory lubricated and should require no attention.

Cleaning and servicing. Access holes are provided for ring cleaning. A clean, soft, non-linting cloth moistened with trichloroethylene or acetone may be used. The epoxy surfaces adjacent to the rings should also be cleaned. Brushes and brush block assemblies should be cleaned with the same solvents.

If brushes are removed, they should be replaced in exactly the same position as before removal. This eliminates the need for re-seating the brush to the ring.

The foregoing abbreviated instructions have been taken from Reference 4. Reference 4 should be consulted for more detailed instructions and information.

REFERENCES

1. Technical Manual - Instructions for Visicorder Oscillograph - Model 1508, Honeywell, Denver Division, 4800 E. Dry Creek Road, Denver, Colorado (Sept 1964)

2. SKF Service Catalog #450, SKF Industries, 1835 Rollins Road, Burlingame, California. (1963)

3. Installation and Service Instruction, ADEL Isodraulic Remote Control Systems, for Manually Operated Systems, (DATA 3-32) ADEL Products Division, General Metals Corp. 10777 Van Owen Street, Burbank, California

4. Instruction Manual for Slip Ring Assembly Model No.6109-12 S/N 102, Lebow Associates, Inc., 14857 W. Eleven Mile Road, Oak Park 37, Michigan.

APPENDIX A

DETERMINATION OF ACCELERATION FIELD

At constant RPM, the acceleration field in the combustion bomb has two components; a vertical component due to the earth's gravitational field, and a radial component which depends on the radius and angular velocity. Hence the acceleration vector always remains in a vertical plane, but both direction and magnitude depend on angular velocity. With the centrifuge at rest, the direction is vertical and downward. At high speed, the direction is nearly horizontal.

The symbols used are defined as follows:

a = radial acceleration/ft radius ($1/sec^2$)

r = radius (in)

N = number in EPUT meter display (RPM/2)

g = acceleration due to gravity ($32.2\ ft/sec^2$)

G_r' = a/g = unit radial load factor per foot radius (1/ft)

G_r = radial load factor = G_r' (r/12 in/ft)(dimensionless)

G = total load factor

ω = angular velocity (radians/sec)

The radial acceleration per foot radius is

$$a = r\omega^2 =((1\ ft)(4\pi N)^2/(60\ sec/min)^2)/(1\ ft)$$

$$= N^2/(4.775)^2\ 1/sec^2$$

and

$$G_r' = a/g = N^2/(4.775)^2 \times 32.2) = N^2/734.5$$

or

$$N = 27.1\ (G_r')^{\frac{1}{2}}$$

At a radius of 35.6 inches (the mid-point of the propellant sample) the radial load factor is

$$G_r = (35.6''/(12\text{ in/ft}))\, G'_r$$
$$= N^2/247.5$$

and $$N = 15.72(G_r)^{\frac{1}{2}}$$

For radial load factors of 10 or more, the contribution of the vertical component is less than one percent. Hence for $G_r > 10$, the acceleration Vector G is assumed to be horizontal and equal to G_r.

i.e.

$$G = N^2/247.5 \qquad \text{(A1)}$$

and

$$N = 15.72\, G^{\frac{1}{2}}$$

This solution is plotted in Figure A1.

For $G_r \leq 10$, the vertical component is taken into account. The solutions are shown in Figures A2 and A3.

Figures A1, A2, and A3 may be used to select the value of N to obtain the desired acceleration. The recorded value of N should be used to calculate G from Equation (A1).

For tests at or below $G_r = 10$, Figure A3 may be used in conjunction with Equation (A1) to calculate G.

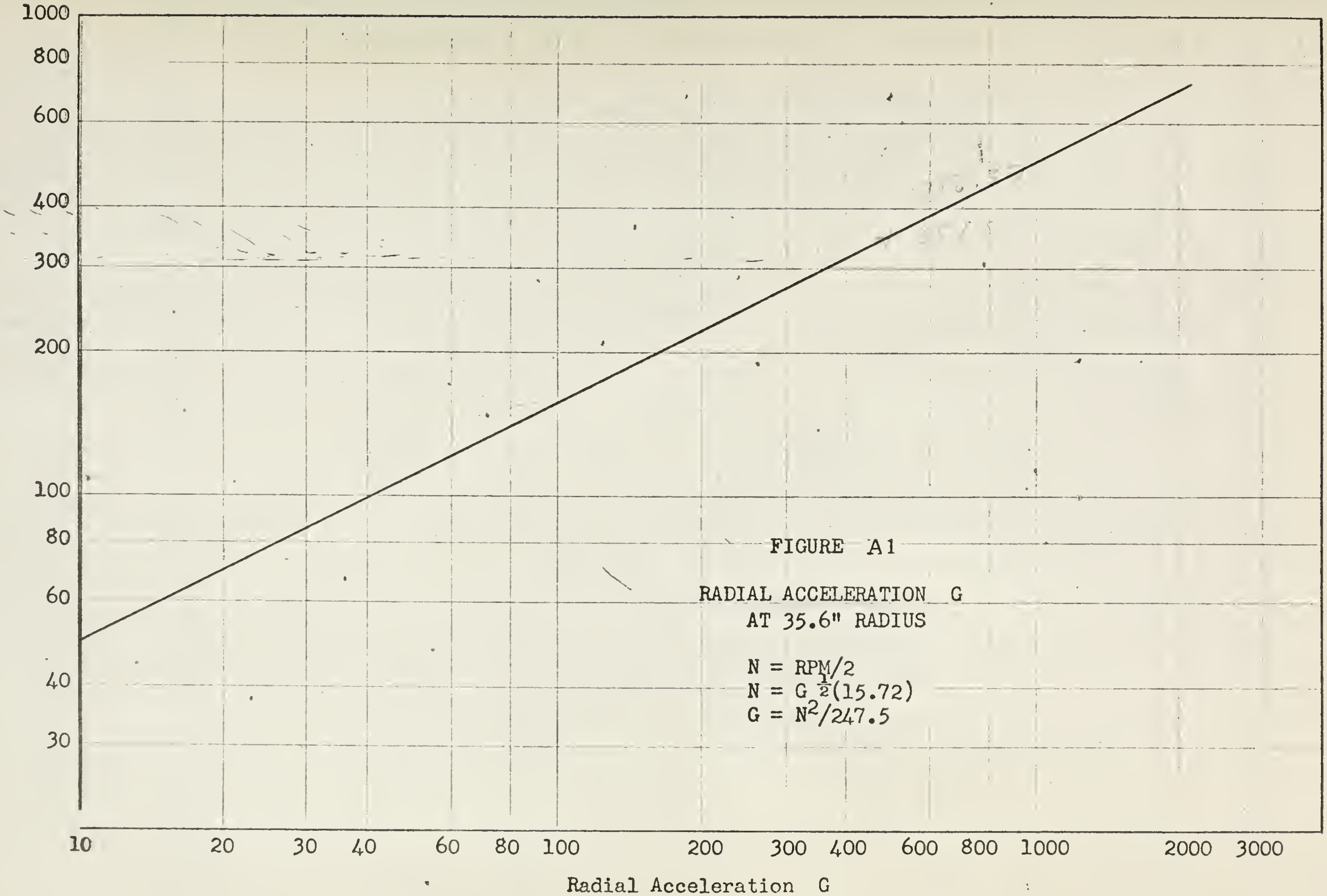


FIGURE A1

RADIAL ACCELERATION G
AT 35.6" RADIUS

$N = RPM/2$
$N = G^{\frac{1}{2}}(15.72)$
$G = N^2/247.5$

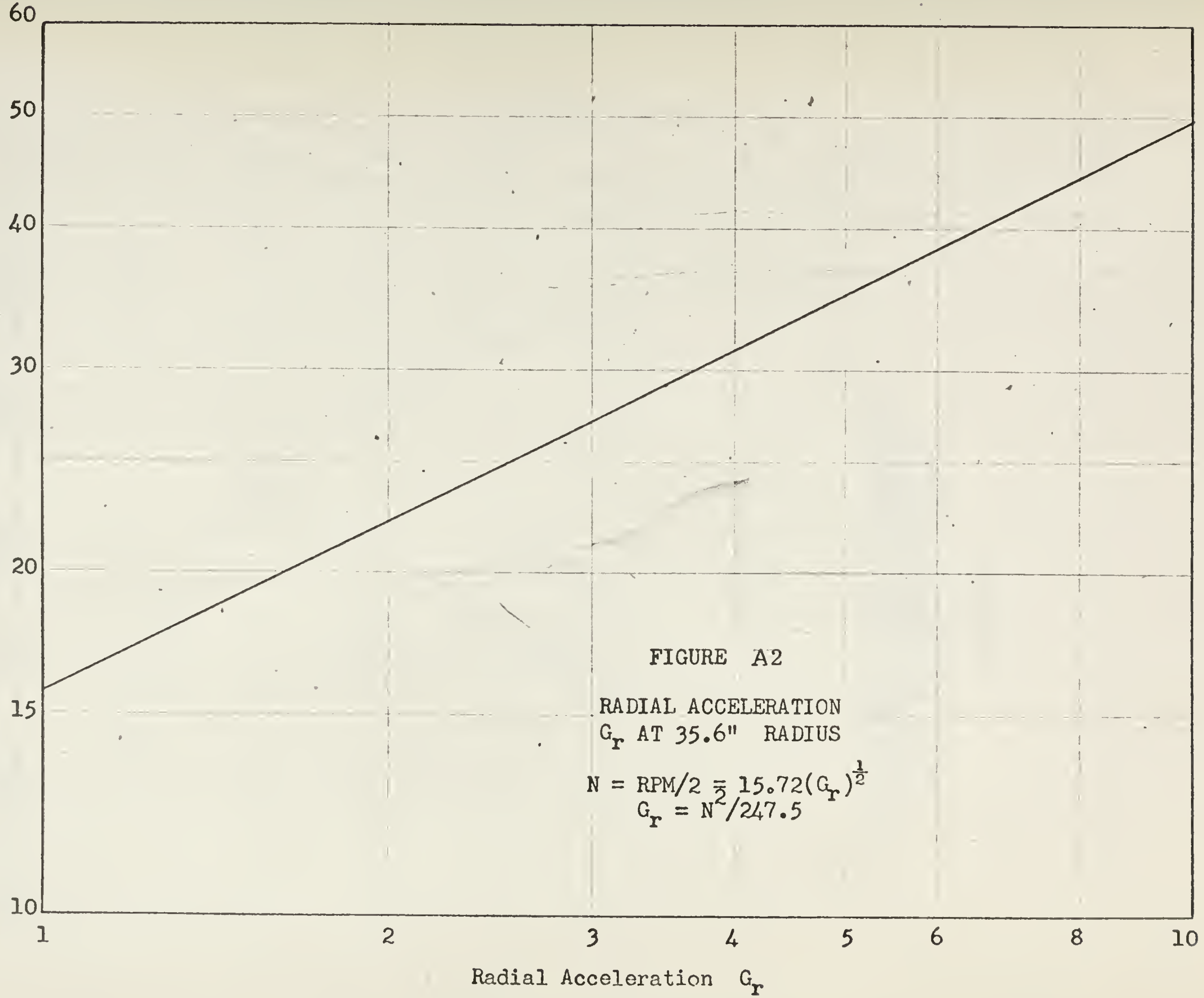


FIGURE A2

RADIAL ACCELERATION
G_r AT 35.6" RADIUS

$N = RPM/2 \mp 15.72(G_r)^{\frac{1}{2}}$
$G_r = N^2/247.5$

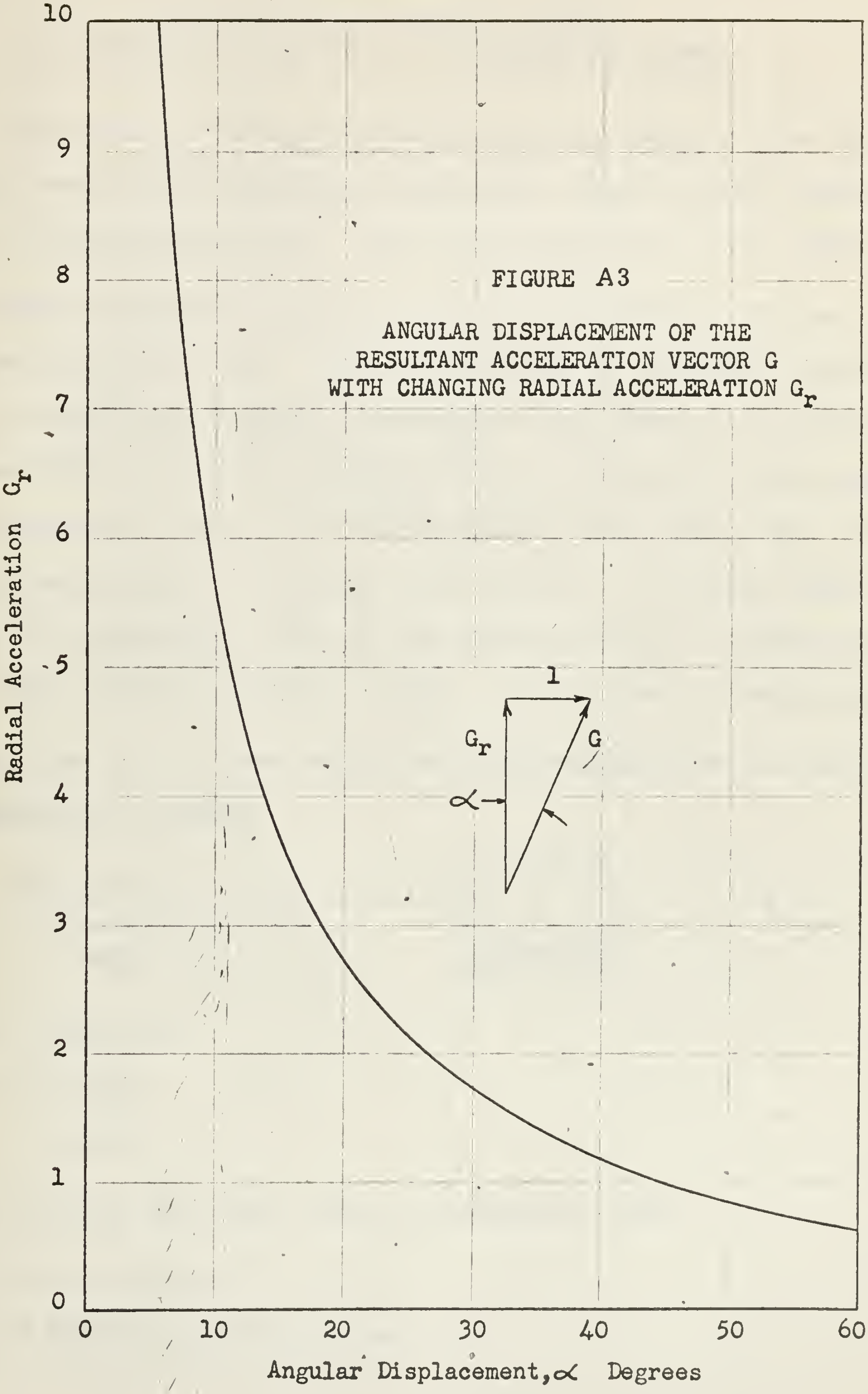
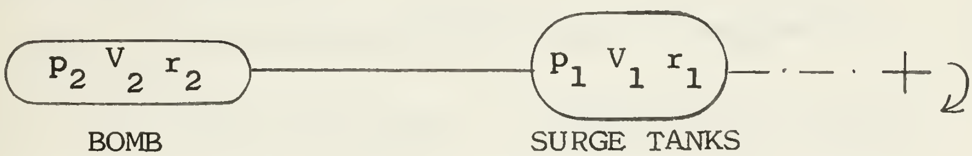


FIGURE A3
ANGULAR DISPLACEMENT OF THE RESULTANT ACCELERATION VECTOR G WITH CHANGING RADIAL ACCELERATION G_r
Radial Acceleration G_r
Angular Displacement, α Degrees
0
1
2
3
4
5
6
7
8
9
10
0
10
20
30
40
50
60
1
G_r
G
α

APPENDIX B

CALCULATION OF PRESSURE CHANGE IN COMBUSTION BOMB DUE TO CENTRIFUGAL FORCE

With the centrifuge in motion, gas confined in the surge tanks, bomb, and connecting tubing will tend to move outward due to centrifugal force. Hence the pressure in the combustion bomb will be higher than the pressure to which the system was charged while at rest. With the centrifuge in motion there will be pressure gradients throughout the field. The problem becomes more tractable, however, if: 1) the gas in the bomb is considered to be at the same pressure throughout and its' mass concentrated at a point, 2) the gas in the two surge tanks is considered to be at the same pressure throughout and its' mass concentrated at a point, 3) the two concentrated masses lie on the same radial and are connected by a thin tube of negligible volume.



let p = pressure

V = volume

r = radius

A = cross sectional area of connecting tube

ρ = gas density

ω = angular velocity 1/sec

R = gas constant for N_2 = 55.2 ftlb/lbm ^oR

T = gas temperature = 528^oR

r_1 = tank radius = 8 in.

r_2 = bomb radius = 35 in.

g = 32.2 ft/sec^2

V_1 = tank volume = 1450 cu.in.

V_2 = bomb volume = 115 cu.in.

p_o = pressure in system with $\omega = 0$

In the connecting tube, where the centrifugal force must be balanced by a pressure gradient,

$$(r\omega^2)(\rho A dr) = A dp \tag{B1}$$

Assuming a perfect gas,

$$\rho = p/RT \tag{B2}$$

combining (B2) and (B1) gives

$$dp/p = (\omega^2/2gRT)2rdr \tag{B3}$$

Assuming T to be constant, and integrating between stations 1 and 2, (B3) becomes

$$p_2/p_1 = \exp\ (r_2^2(1-r_1^2/r_2^2)\omega^2/g2RT) = \exp(k\omega^2/g) \tag{B4}$$

where

$$k = r_2^2(1-r_1^2/r_2^2)/2RT = 1.380\times10^{-4}\ \text{ft-lbm/lb}$$

Conservation of mass, when combined with (B2) gives

$$p_o(V_1 + V_2 + A(r_2-r_1)) = p_1V_1 + p_2V_2 + \int_{r_1}^{r_2} A\rho dr \tag{B5}$$

Neglecting the small amount of gas in the connecting tube, and rearranging (B5) gives

$$p_2/p_o = (V_2 + V_1)/(V_2 + V_1p_1/p_2) \tag{B6}$$

or

$$p_2/p_o = 1/(1-(1-p_1/p_2)/(1 + V_2/V_1)) \tag{B7}$$

From (B4), $(p_1/p_2)\min = [\exp(-k\omega^2/g)]\min$, which occurs at the maximum speed of 155 rad/sec.

$$(k\omega^2/g)\max = 0.1032$$

and

$$(p_1/p_2)\min = 0.92$$

Hence $$((1-p_1/p_2)/(1 + V_2/V_1))\max = 0.0742$$

and

$$p_2/p_o \simeq 1 + (1-p_1/p_2)/(1 + V_2/V_1) \qquad \text{(B8)}$$

Letting $p_2 = p_o + \Delta p$, and combining (B4) with (B8),

$$\Delta p/p_o \quad (1-\exp(-k\omega^2/g))/(1 + V_2/V_1) \qquad \text{(B9)}$$

$$\exp(-k\omega^2/g) = 1-k\omega^2/g + (k\omega^2/g)^2/2 - \cdots$$

$$\simeq 1-k\omega^2/g \text{ since } k\omega^2/g \leq 0.1032$$

Hence $$\Delta p/p_o \simeq (k\omega^2/g)/(1 + V_2/V_1) = 1.284\text{x}10^{-4}\ \omega^2/g \qquad \text{(B10)}$$

The curves shown in Figure B1 were derived from Equation (B10).

With only one surge tank connected to the bomb, the pressure rise due to centrifugal force is not as large. Using the same solution as before,

$$\Delta p/p_o \simeq (k\omega^2/g)/(1 + V_2/V_1)$$

where $$k = r_2^2\,(1 - r_1^2/r_2^2)/2RT$$

and $$R = 55.2 \text{ ft lb/lbm } ^oR$$

$$T = 528\ ^oR$$

$$r_1 = 8 \text{ in.}$$

$$r_2 = 35 \text{ in.}$$

$$V_1 = 725 \text{ cu. in.}$$

$$V_2 = 115 \text{ cu. in.}$$

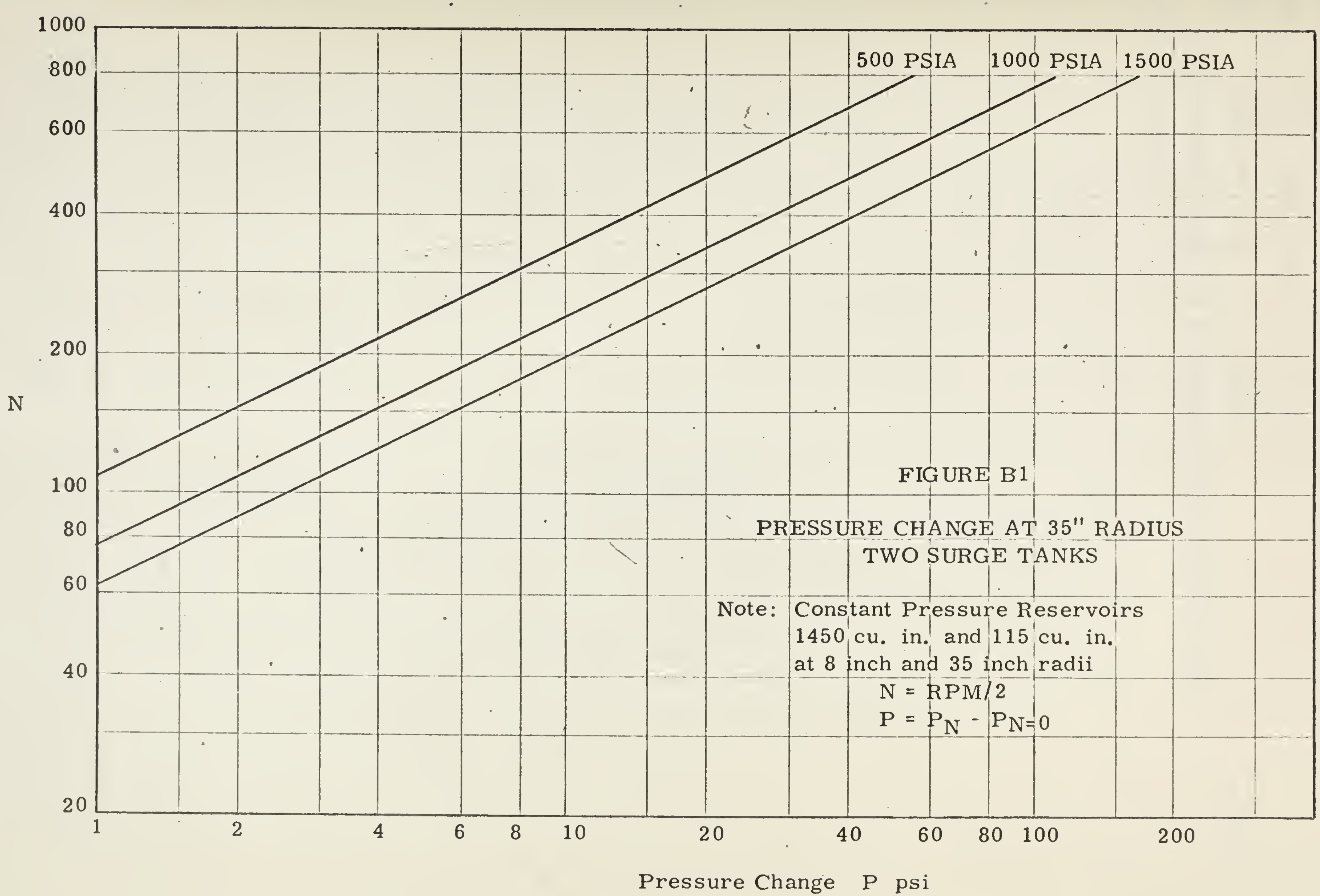


FIGURE B1

PRESSURE CHANGE AT 35" RADIUS
TWO SURGE TANKS

Hence $\quad k = 1.380 \times 10^{-4}$ ft lbm/lb

and $\quad \Delta p/p_o \simeq 1.191 \times 10^{-4}\ \omega^2/g \qquad$ (B11)

The curves shown in Figure B2 were derived from Equation (B11).

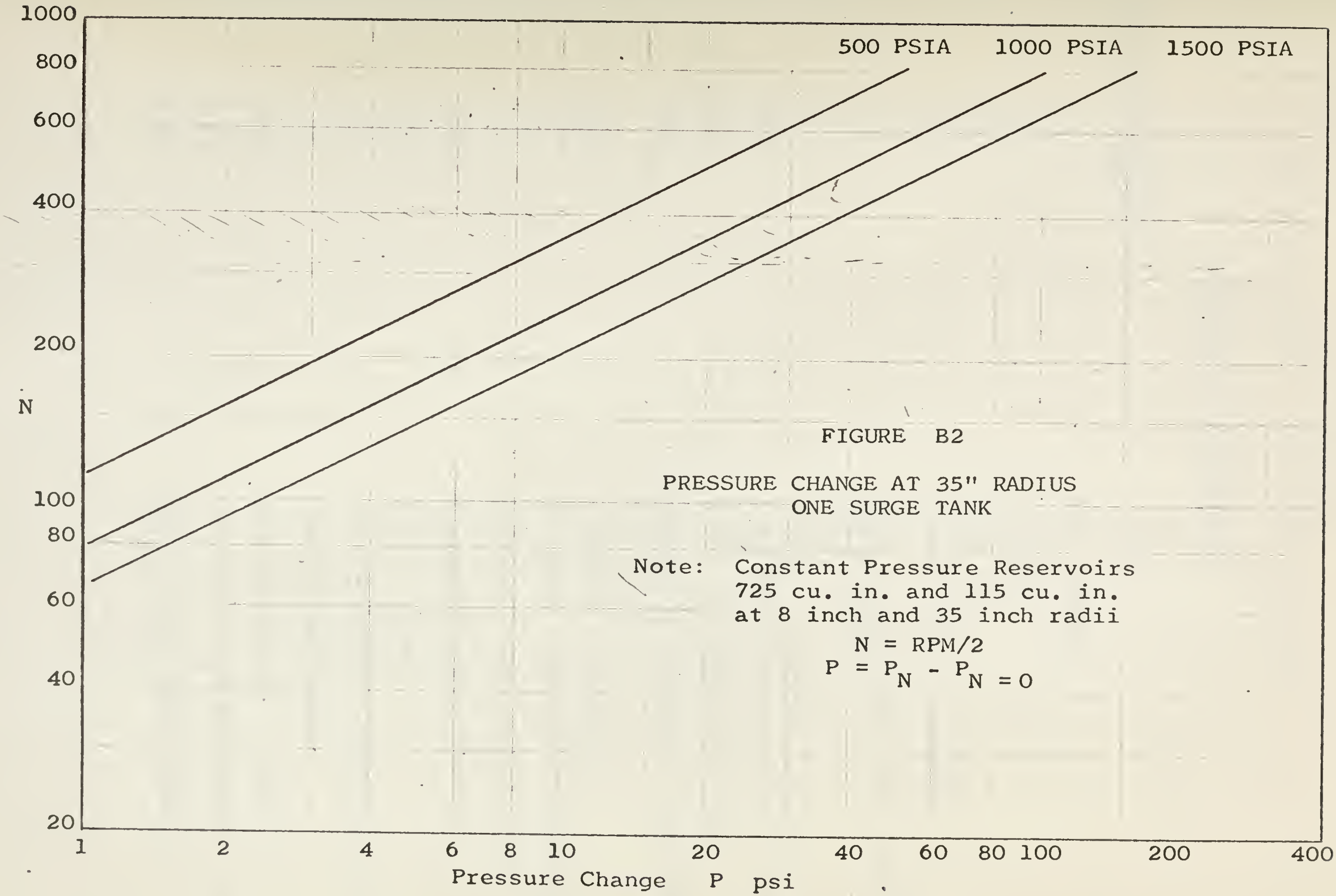


FIGURE B2

PRESSURE CHANGE AT 35" RADIUS
ONE SURGE TANK

APPENDIX C

LIST OF DRAWINGS

76 inch Diameter Centrifuge Facility

U. S. Naval Postgraduate School
Monterey, California

Aeronautics Dept. Drawing number	Title
1131B	Bomb Assembly
1132A	Retainer
1134A	Bomb Plug
1135A	Strand Holder Assembly
1136	Bomb attach detail (not issued) see Dwg 1162
1137A	Collar
1138B	Assy, Centrifuge and Power Plant
1139	Details - Engine Sub-base
1140	Details - Sub-base - Centrifuge (2 sheets)
1141A	Coupling - Shear Pin
1142	Retainer - Shear Pin Coupling
1143A	Adaptor - Shear Pin Coupling
1144A	Bracket - Brake Mounting
1145A	Disc - Brake
1146	Seal Retainer - Lower Bearing
1147	Spacer—Main Shaft
1148	Bearing Housing - Lower
1149	Details - Bearing Support Legs
1150	Drill Jig - Upper Bearing Support Legs

Aeronautics Dept. Drawing Number	Title
1151A	Upper Bearing Assembly
1152	Mounting Plate - Gear Box
1153D	Vertical Shaft - Centrifuge
1154	Sub-base Details - Centrifuge
1155	Seal Retainer - Lower Bfg.
1156	Assy Details - Lower Bearing
1157	Alignment Disc - Upper Bearing Housing
1158B	Pivot Pin - Centrifuge Arm
1159	Console (2 sheets)
1160	Instrument Layout - Console
1161	Brake Pedal (3 sheets)
1162	Centrifuge Rotor Details (6 sheets)
1163	Throttle Linkage
1164	U. Sprocket
1165	L. Sprocket
1166	Rotor Detail
1201	Balancing Stand
1168	Bomb Plug MK II
1169	Temp Sensor Adaptor
1170	Jig

APPENDIX D

STRESS ANALYSIS

The stress analysis is divided into two major parts. Part I is concerned with the centrifuge shaft, support structure, brake system, and drive system. Part II is concerned with the rotor assembly. This includes the combustion bomb and its attachment to the arms, the arms themselves, counterweight attachment, surge tanks, tubes and fittings.

PART I

General

The worst loading condition on the centrifuge structure is assumed to be the load resulting from the loss of an 18 pound test package (in this case the bomb) at a speed of 1450 RPM. Assuming the structural damping to be very small, the maximum peak load applied to the centrifuge shaft would be 72 kips, while the "static" load would be 36 kips. The cyclic frequency of the 36 kip loading (24 cps) was assumed to be sufficiently below the resonant frequency of the structure so as to make a static analysis using the peak load sufficiently accurate. Figure D1 shows the peak loading condition.

The centrifuge base structure and shaft are designed to withstand the loads shown in Fig. D1 without failure. Symbols and strength of materials values given in MIL-HDBK-5 (Ref. 1) have been used. Additional symbols are:

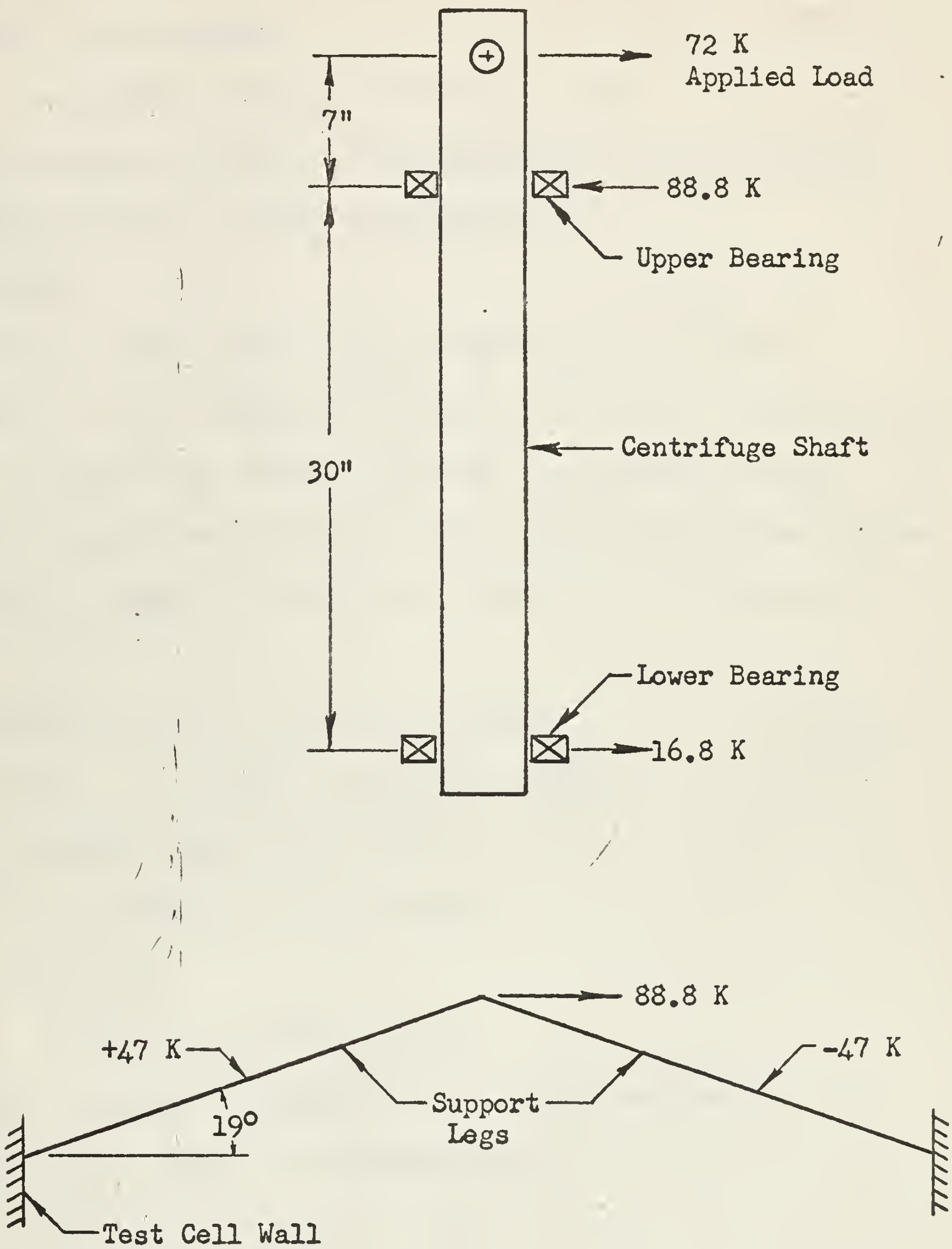


FIGURE D1

PEAK LOADING CONDITION
CENTIFUGE STRUCTURE

D = outside diameter

d = inside diameter

k = kips = kilopounds

in-k = inch-kips (bending moment) = kin

Drawing numbers refer to the Department of Aeronautics drawing number series listed in Appendix C.

<u>Pin</u>, <u>Dwg</u> <u>1158B</u>

Material: Alloy steel ($R_b = 95 \geqslant F_{tu} - 100$ ksi)

Assume: 1) 36k applied to each end of pin concentrated at midpoint of bearing surface between rotor arm and pin. 2) Forces on shaft applied uniformly over shaft-pin contact area.

<u>Bearing on shaft</u>. 36k force, 3/8 thick shaft wall, 1½ dia. pin,

f_{br} =(36k)/(1.50" x .375") = 64 ksi

F_{bru} = 140 ksi $\qquad f_{br}/F_{br}$ = 64/140 = <u>.457</u>

<u>Shear</u>. Shear Area = 1.76 in^2

f_s = 36k/1.76 = 20.5ksi

Assume F_{su} = 50 ksi

f_s/F_{su} = 20.5/50 = <u>.41</u>

<u>Bending</u>. Assuming concentrated loads applied at mid points of bearing areas, the moment arm is

½ (1 1/8" + 3/8") = 3/4"

Bending moment = (3/4")(36)k = 27 in-k

Z pin = .331 in^3 $\qquad f_b$ = M/Z = 27 kin/.331 = 81.7 ksi

F_b = 160 ksi $\qquad f_b/F_b$ = 81.7/160 = <u>.51</u>

$f_s/F_s + f_b/F_b$ = .41 + .51 = <u>.92</u>

Shaft DWG 1153D

Material: 4130 steel (normalized)

$F_{tu} = 90$ ksi

$F_{su} = 55$ ksi

See Figure D2 for a sketch of the shaft and the design loading condition. The capital letters below refer to shaft locations shown in Figure D2.

At A and C. $D = 4.5$ in, $d = 3.75$ in

$Z = (\pi/32)(D^4 - d^4)/D = 4.63\ in^3$

At A. $f_b = (72k \times 5.0625")/(4.63\ in^3) = 78.5$ ksi

$D/t = 4.5/.375 = 12 \Rightarrow F_b = 118$ ksi, f_b/F_b

$f_b/F_b = 78.5/118 = \underline{.665}$

Shear Area $= \pi(D^2 - d^2)/4 = 4.86\ in^2$

$f_s = 72/4.86 = 14.8$ ksi

$f_s/F_s = 14.8/55 = \underline{.27}$

$f_s/F_s = f_b/F_b = .27 + .665 = \underline{.94}$

At B. $D = 4.72$ in, $d = 3.75$ in, $Z = 6.2\ in^3$

$f_b = M/Z = 485\ in\text{-}k/6.2\ in^3 = 78.3$ ksi

$D/t = 4.72\ in/.48\ in. = 9.84 > F_b = 122$ ksi

$f_b/F_b = 78.3/122 = \underline{.642}$

Shear Area $\pi(D^2 - d^2)/4 = 6.37\ in^2$

$f_s = 72k/6.37\ in^2 = 11.3$ ksi

$f_s/F_{su} = 11.3/55 = \underline{.20}$

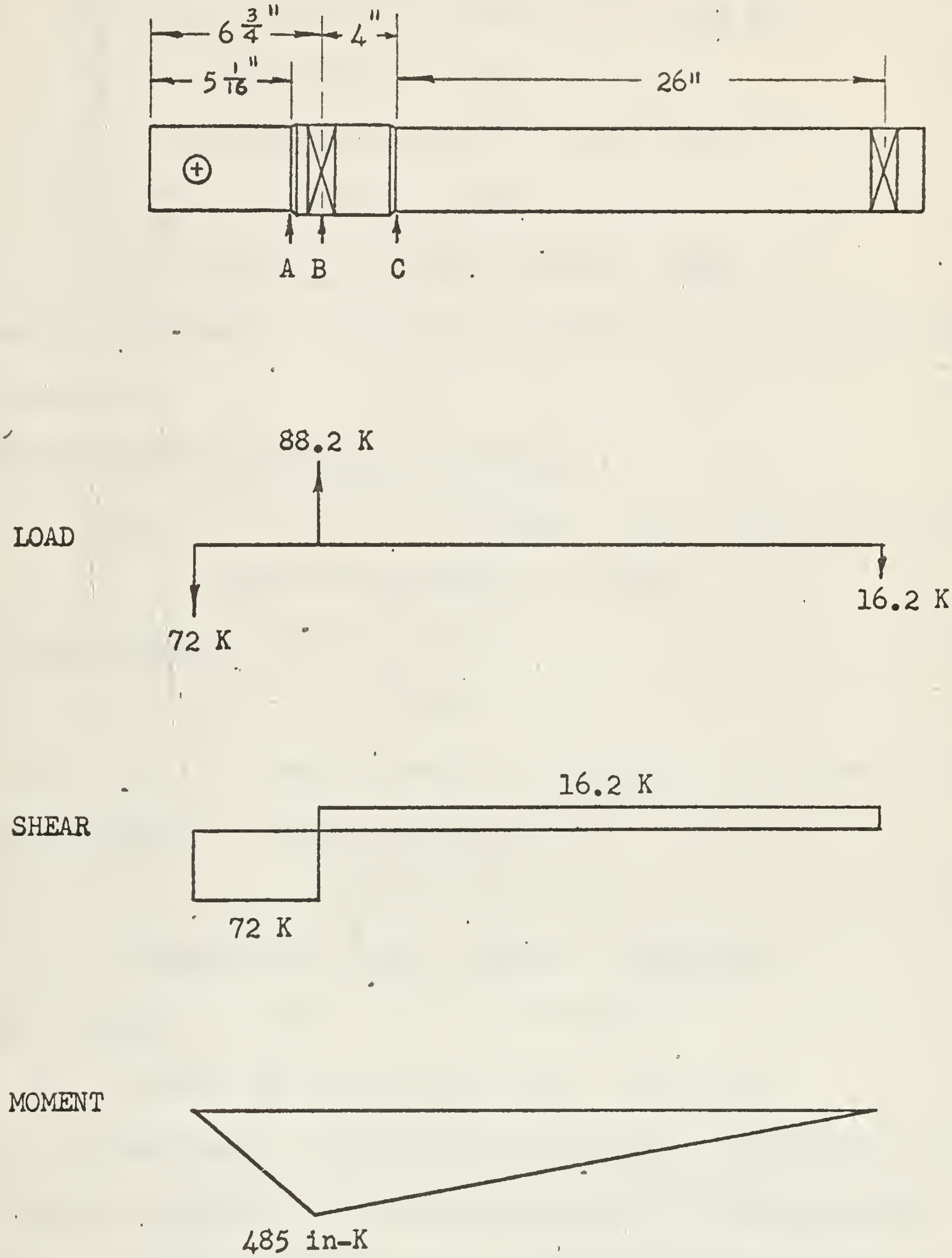


FIGURE D2

PEAK LOADING CONDITION
CENTRIFUGE SHAFT

$f_b/F_b + f_s/F_s = .642 + .20 = \underline{.84}$

<u>At C.</u> $Z = 4.63\ in^3$, $M = (26/30)(485\ in\text{-}k) = 420\ in\text{-}k$

$f_b = M/Z = 420\ in\text{-}k/4.63\ in^3 = 90.8\ ksi$

$f_b/F_b = 90.8/118 = \underline{.768}$

$f_s = 16.2\ kips/4.86\ in^2 = 3.33\ ksi$

$f_s/F_{su} = 3.33/55 = .061$

$f_b/F_b + f_s/F_{su} = .768 + .061 = \underline{.829}$

Hence the shaft is critical at point A, just above the upper bearing.

<u>Upper Bearing Housing DWG 1151A</u>

Material: 6061-T6 Aluminum $F_{ty} = 35ksi$ $F_{bry} = 50ksi$

MAX RADIAL LOAD = 88k

<u>Race: Housing</u> Dia = 8.46 in

h = 2 in

$f_{br} = F/A = (88k)/(8.46\ in. \times 2\ in.) = 5.2\ ksi$

<u>Housing: Support</u> Dia = 13.5 in

h = .25 in

$f_{br} = (88k)/(13.5\ in \times .25\ in) = \underline{26.1}\ ksi$

<u>Housing: Bolts</u> ½" dia, h = 1", 6 bolts

$f_{br} = (88k)/(\frac{1}{2}\ in \times 1\ in \times 6) = 29.3\ ksi$

Load is transmitted to the bearing housing)P/N 1151) to the Upper Bearing Support (P/N 1140-1) via a ¼" high shoulder, and the load is then carried to the walls of the test cell by the support legs (P/N 1138-4). See Figure D1. The upper support

is designed so that the point of intersection of the leg center-lines is at the center of the upper bearing.

Support Legs DWG 1138B

Material: 3" std. (SCHED 40) steel pipe

Metal Area = 2.23 in^2 Z = 1.724 in^3 ρ = 1.164 in

Mac load: $\pm$ 47 kips L = 98½ inches

Assume no eccentricity, ℄ end fixity C = 2

$L'/\rho = (98.5")/(1.164 \text{ in} \times 1.414) = 59.8$

$F_c = \pi^2 E/(L'/\rho)^2 = (9.85 \times 29 \times 10^6 \text{psi})/(3.58 \cdot 10^3) = 80$ ksi

F_{co} = 36 ksi $\Rightarrow$ the Euler curve does not apply and the column is short.

$f_c = 47 \text{ k}/2.23 \text{ in}^2 = 21.1$ ksi

$f_c/F_{co} = 21.1/36 = \underline{.586}$

$B^2 = F_{co}/F_c = 36/80 = .45$, $\underline{B = .67}$

Plotting this point on Fig. 1.6.3.2 in MIL-HDBK-5 shows the leg to be safe.

The support legs are attached to 3½" std pipe sleeves at each end by means of six ½" dia. heat-treated shoulder screws.

$R_c = 32$ (F_{tu} = 144 ksi)

f_{br} = (47 kips)/(12 x .5" dia. x .216") = 36.3 ksi

F_{bru} = 90 ksi $f_{br}/F_{bru} = 36.3/90 = \underline{.403}$

$f_s = (47 \text{ kips})/(12 \times .1963 \text{ in}^2) = 20$ ksi

$F_{su} > 75$ ksi, $f_s/F_{su} < 20/75 = \underline{.267}$

The maximum bending moment arm in any one bolt would be

½(.216" - .226") + (3.548 - 3.5) = .269"

The resultant bending moment is 47k (.269 in.)/12 bolts

$= 1.055$ in-k

$f_b = M/Z = 1.055$ ink/.01228 in^3 $= 85.8$ ksi

$F_b = 230$ ksi, $f_b/F_b = \underline{.323}$

$f_s/F_{su} + f_b/F_b = .267 + .323 = \underline{.590}$

<u>Upper Sleeves DWG 1140-2A</u>

Material: 3½" std. steel pipe

The upper sleeves are welded to the Upper Bearing Support. In compression, approximately one-half of the 44.4 kip horizontal load is transmitted through the 13.5 inch long welded joint on the underside of the support. The shear area of the joint is at least 3/16" x 13.5" = 2.53 in^2

$f_s = 22.2$ k/2.53 in^2 $= 8.78$ ksi

$f_s/F_s = 8.78/35$ ksi $= \underline{.35}$

The other half of the load goes into the upper sleeve via direct compression through a 6 inch long x 3/16" wide weldment on the side of the Upper Bearing Support.

In tension, assuming a uniform stress in the sleeve, the weldment forces on the side of the Upper Bearing Support are

Tension: $22.2\text{k}(\cos 19^\circ) = 21.2\text{k}$

Shear: $22.2\text{k}(\sin 19^\circ) = 7.22\text{k}$

$f_t = (21.2 \text{ kips})/(6 \text{ in} \times 3/16 \text{ in}) = 18.85$ ksi

$f_s = (7.22\text{k})/(6 \text{ in} \times 3/16 \text{ in}) = 6.42$ ksi

$f_t/F_{ty} + f_s/F_{su} = 18.85/36 + 6.42/35 = .524 + .184 = \underline{.708}$

The area of the weldment on the underside of the Upper Bearing Support is 2.53 in^2, hence the stress there is not critical.

Lower Sleeves DWG 1138B

Material: 3½" Std. Steel Pipe

The lower sleeves are welded to 3/4 inch thick steel plates at an angle of 42°. The length of the joint L is

$$L = (\pi)/((3.75^2 + 5.5^2)/2)^{\frac{1}{2}} = 14.7''$$

The shear force to be transmitted by the weld metal is $47k\,(\cos 42^{\circ}) = 34.4$ kips

$f_s = 34.4k/(14.7'' \times .1875'') = 12.5$ ksi

$f_s/F_{su} = 12.5/35 = \underline{.358}$

The tensile force to be transmitted by the joint is

$47k(\sin 42^{\circ}) = 31.4$ kips

$f_t = 31.4/(14.7'' \times .1875) = 11.4$ ksi

$f_t/F_{ty} = 11.4/36 = \underline{.317}$

$f_s/F_{su} + f_t/F_{ty} = .358 + .317 = \underline{.675}$

Wall Attachment DWG 1138B

MAX LOAD = 47 kips axial compression/tension in support leg.

4 1" DIA C1018 steel stud bolts, 1" dia. in shear $\Rightarrow .995\ in^2$

1 - 14 THD, Minor Dia. .907 in $\Rightarrow$ Area in tension = $.647\ in^2$

Angle between leg and wall = 42°

Shear force = $47k\,(\cos 42^{\circ}) = 34.4$ kips

Tensile force = $47k(\sin 42^{\circ}) = 31.4$ kips

$f_s = 34.4k/(4 \times .995\ in^2) = 8.64$ ksi $\quad f_s/F_{su} = 8.64/35 = \underline{.247}$

$f_t = 31.4k/(4 \times .647\ in^2) = 12.12$ ksi $\quad f_t/F_{ty} = 12.12/36 = \underline{.337}$

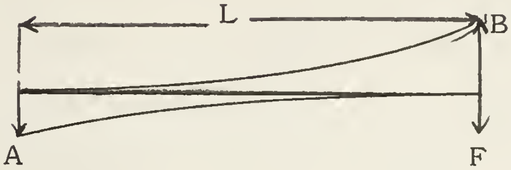
$f_s/F_{su} + f_t/F_{ty} = .247 + .337 = \underline{.584}$

Bearing of bolts on wall.

Shear force = 34.4k/4 = 8.6 kips per bolt

Assume applied force varies as the square of the distance,

i.e.,



F: $AL/3 + F = BL/3$, $-A + B = 3F/L$, $-2B = -9F/L$

M: $-AL^2/12 - LF + BL^2/4 = 0$, $A - 3B = -12F/L$, $B = 9F/2L$

$F = 8.6k$, $L = 12''$, $B = 9(8.6)/(2\times(12)) = 3.23$ ksi

<u>Lower Bearing Housing DWG 1148</u>

Material: Mild steel

The lower bearing, an SKF Number 1222, is a slip fit in the Bearing Housing, except for the last ¼ inch which is a push fit. The Bearing housing itself is a slip fit in the welded base structure. Radial loads are transmitted directly through the housing into the top flange of the base structure.

<u>Base Assembly DWG 1140A</u>

Material: Standard Rolled Steel Shapes

The Base Assembly is constructed from 12" x 12" WF65# and 12" x 25# std. channel. It is anchored to the thrust pad rails in the floor by two 3/4" anchor bolts in each of the four corners.

<u>Bearings</u>

The Upper Bearing is an SKF Number 22224C spherical roller bearing (self-aligning). The bearing is rated for a radial load of 32.2 kips for 5000 hrs at 100 RPM, and 8.35 kips for 30,000 hrs at 1500 RPM. (Ref. 2).

During normal operating conditions the rotor should be in static balance within a quarter of a pound. At 2000g the resultant radial load applied to the Pin (DWG 1158B) should be less than 500 pounds. The radial load carried by the upper bearing would then be less than 617 pounds.

The Lower Bearing is an SKF Number 1222 self-aligning ball bearing with two rows of balls. It is rated for a radial load of 4.89 kips for 5000 hrs at 100 RPM, and 1.09 kips for 30,000 hrs at 1500 RPM. With a ¼ pound static imbalance, the radial load on the lower bearing would be 117 pounds. The thrust load to be carried by the lower bearing is

Rotor & Pin	150 lb
Shaft	60 lb
Slip-Ring Assy ...	50 lb
Brake Disc	15 lb
Total	275 lb

Let R = radial load

T = thrust load

RE = R + 3T = equivalent load (Ref 2)

Bearing No.	Normal Loads, Kips			Max Loads, Kips			Permissible Load
	R	T	RE	R	T	RE	30,000 hrs, 1500 RPM
22224C	.617	0	.617	88.8	0	88.8	8.35
1222	.117	.275	.942	16.8	.275	17.6	1.09

Hence the bearings are adequate for normal loads. Very high loads, such as those resulting from loss of the test packages, would be cause for inspection of the bearings and possible replacement.

Brake System DWGS 1144A, 1145A, 1161

The stopping torque which could be applied by the Triumph TR-3 disc brake is calculated as follows.

The diameter of the master cylinder piston is 7/8 inches, hence the area is 0.6013 in^2. The mechanical advantage of the brake pedal (DWG 1161) is 3.5 and the assumed maximum force applied to the pedal is 175 lb.

$$\text{Piston force} = (3.5)(175 \text{ lb}) = 612 \text{ lb}$$

$$\text{Hyd. Fluid Pressure} = 612 \text{ lb}/0.6013 = 1020 \text{ psig}$$

The diamter of the brake (slave piston) is 2 1/8".

$$\text{Slave Piston area} = 3.55 \text{ in}^2$$

$$\text{Force on brake puck} = (3.55 \text{ in}^2)(1020 \text{ psig}) = 3{,}620 \text{ lb}$$

For a coefficient of friction = .4, the stopping force P applied to the brake disc would be

$$P = (.4)(3620 \text{ lb})(2 \text{ pucks}) = 2900 \text{ lb}$$

The center of the puck assembly is at a radius of 4¼ inches. The stopping torque T is

$$T = (4\tfrac{1}{4} \text{ in})(2900 \text{ lb}) = 12{,}300 \text{ in-lb} = 1{,}025 \text{ ft-lb.}$$

Brake Mounting Bracket DWG 1144A

Material: Mild Steel

The brake caliper is secured to the mounting bracket by

two 3/8" - 16 bolts. Consider a radial from the shaft center line through the center of the brake puck. Let this radial coincide with the x-axis. The y-axis is perpendicular to the x-axis and the shaft centerline and through their intersection. Then the coordinates of the brake puck and mounting bolts are

Brake puck: $x = 4.25$ in

$y = 0$

Mounting bolts: $x = 2.937$ in

$y = \pm 1.75$ in

Let P = puck force = 2.9 k

B = shear force on each bolt

FY: 2 By + P = 0, -By = P/2 = 1.45 k

M (about puck center): 2 By x (4.25 in - 2.937 in)=(3.5 in)x(Bx)

$Bx = (2.9k)x(1.313\text{ in})/(3.5\text{ in}) = 1.087k$

$B^2 = (Bx)^2 + (By)^2 = 2.11 + 1.182 = 3.29\ k^2$

$B = 1.815$ k

The mounting bolts have a minor diameter of 0.2983 in. Hence the shear area is 0.07 in^2.

$$f_s = 1.815\text{ k}/.07\text{ in}^2 = 25.9\text{ ksi}$$

$$f_s/F_{su} = 25.9/35 = \underline{.74}$$

The maximum bending moment on each 1 in dia. x 1 3/8 in long mounting stud is

$M = (1.815\text{ k}) \times 1.375\text{ in} = 2.495$ in-k

$f_b = M/Z = (2.495\text{ in-k})/.0981\text{ in}^3 = 26.45$ ksi

$f_b/F_{ty} = 26.45/36 = \underline{.735}$

Brake disc DWG 1145A

Material: Mild Steel

The shear force at the surface of the 4½ inch diameter shaft is

$$12{,}300 \text{ in-lb}/2\tfrac{1}{4} \text{ in} = 5{,}470 \text{ lb}$$

For an AN6 bolt in double shear,

$$f_s = (5470 \text{ lb}/2) \times .1105 \text{ in}^2 = 24.8 \text{ ksi}$$

$$f_s/F_{su} = 24.8/75 = \underline{.331}$$

Bearing on the ¼ inch thick sleeve of the brake disc:

$$f_{br} = 5470 \text{ lb}/(2 \times .25 \text{ in} \times .375 \text{ in}) = 29.2 \text{ ksi}$$

$$f_{br}/F_{bru} = 29.2/90 = \underline{.325}$$

Brake line.

The line connecting the master cylinder to the brake caliper is ¼" O.D. x .030" wall annealed copper tubing. The hoop stress at 1020 psig is

$$f_t = (1020 \text{ psig} \times .125")/ .030" = 4.25 \text{ ksi}$$

$$F_{tu} = 36 \text{ ksi for soft-annealed copper wire (Ref. 3)}$$

$$f_t/F_{tu} = 4.25/36 = \underline{.118}$$

Shear Coupling DWGS 1141A, 1142, 1143A

Material: Mild steel

Four 1/8 in. dia. brazing rod shear pins on 3.0 in. bolt circle.

Single shear force* = 570 lb per pin by lab test.

Shear torque = (1.5")(570 lb)(4 pins) = 4320 in-lb

= 285 ft-lb

Shear H.P. = (285 ft lb x 1450 RPM)/(5260 ft lb RPM/HP)

= 78.6 H.P. at 1450 RPM

*Ultimate shear force

Flexible Coupling DWGS 1156A, 1164, 1165

Components: Two 6.37" pitch diameter steel sprockets.

20" length of double strand #40-2 standard roller chain.

The maximum torque to be transmitted is 4320 in-lb. The upper sprocket is attached to the shaft by four ¼-28 steel bolts on a 3 3/4 dia. bolt circle.

Shear force on each bolt - 3420 in-lb/(3 3/4" x 2) = 457 lb.

Shear area of minor dia = 0.0326 in^2 (Ref. 3)

$$f_s = 457 \text{ lb}/.0326 \text{ in}^2 = 14 \text{ ksi}$$

$$f_s/F_{su} = 14/35 = \underline{.4}$$

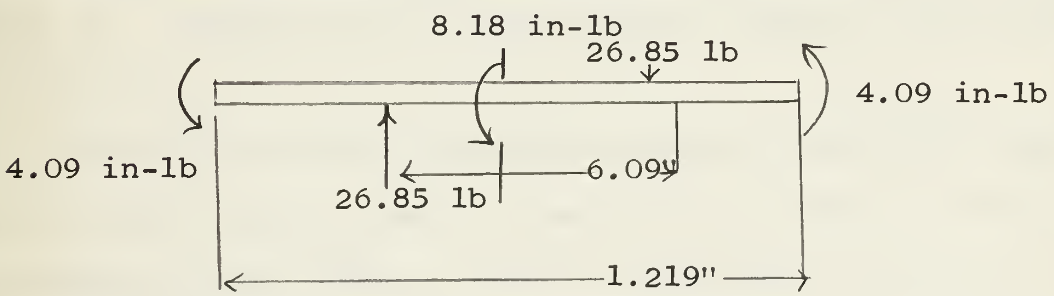
The total driving force P transmitted by the roller chain is

$$P = (3420 \text{ in-lb} \times 2)/6.37'' = 1074 \text{ lb}$$

The sprockets have 40 teeth, so the force carried by each pin of the chain is

$$1074 \text{ lb}/40 = 26.85 \text{ lb}$$

Consider a single pin (see Ref. 4 for chain dimensions)



The maximum bending moment in the pin is 4.09 in-lb

$$Z = \pi D^3/32 = \pi(.156)^3/32 = 3.73 \cdot 10^{-4} \text{ in}^3$$

$$f_b = M/Z = 4.09 \text{ in lb}/3.73 \cdot 10^{-4} \text{ in}^3 = 10.96 \text{ ksi}$$

The roller pins are hardened steel, so f_b = 10.96 ksi should be a sufficiently low stress.

Using the chain and sprocket dimensions in Ref. 4, the maximum permissible angular misalignment θ is

$$\mathrm{Tan}\ \theta = 4\ (W - T)/P.D.$$

where W = roller width = 5/16"

T = sprocket thickness = .284" max

P.D. = pitch dia. = 6.37

$$\mathrm{Tan}\ \theta = 4(.3125 - .284)/6.37$$

$$= 4\ (.0285)/6.37 = .01764$$

$$\theta = (.01764\ \mathrm{rad})(57.3^{\circ}/\mathrm{rad}) = 1.01^{\circ}$$

PART II

General

The maximum static loading condition on the rotor assembly occurs at the design capacity of 36,000 g-pounds at 1450 RPM. That is, at a speed of 1450 RPM, the 18 pound combustion bomb is in a centrifugal force field of 2000 g.

The rotor assembly is designed to one-half the yield point based on strength of materials values given in MIL-HDBK-5 (Ref. 1). Symbols are defined in Ref. 1 and at the beginning of Part I.

Loss of the 18 lb. combustion bomb at 1450 RPM would lead to the peak loading condition described in Part I. Hence yielding but not failure could be expected in critical regions of the rotor arms of the counter-weight end.

Bomb Retainer DWGS 1133A, 1162

Material: 2024-T4 Aluminum tube

$F_{tu} = 60$ ksi $\qquad F_{ty} = 40$ ksi

$F_{cy} = 38$ ksi $\qquad F_{su} = 38$ ksi

$F_{bry} = 74$ ksi $\qquad (e/D = 2)$

The Bomb Retainer is designed to carry a load of 36 kips.

<u>Shear in 4 15/16" - 8N class 2B threads</u>.

The thread area in shear is

$$a = \pi DL/2$$

$$D = \text{pitch diameter} = 4.86 \text{ in}$$

$$L = \text{thread engagement length} = 1''$$

$$A = \pi \times 4.86 \text{ in} \times \tfrac{1}{2} = 7.63 \text{ in}^2$$

$$f_s = 36\text{k}/7.63 \text{ in}^2 = 4.72 \text{ ksi}$$

$$f_s/F_{su} = 4.72/38 = \underline{.124}$$

<u>Hoop stress</u>

Assuming a 45° thread and no friction, an axial load of 36 kips gives rise to a radial stress p where

$$p = 36\text{k}/(\pi \times 4.86 \text{ in} \times 1 \text{ in}) \quad 2.36 \text{ ksi}$$

The resultant hoop sress f_t is

$$f_t = pr/t = 2.36 \text{ ksi} \times 2.43 \text{ in}/ .25 \text{ in} = 23.8 \text{ ksi}$$

$$f_t/F_{ty} = 23.8/40 = \underline{0.595}$$

The increase in pitch diameter due to the hoop stress is

$$d = Df_s/E = 4.86 \times 23.8/10^4 = .0187 \text{ in}$$

<u>Compressive stress</u>

The compressive stress in the cylindrical retainer is

$$f_c = P/A$$

where $A = \pi((5.5\ \text{in})^2 - (4.75\ \text{in})^2)/4 = 6.04\ \text{in}^2$

$$f_c = 36\text{k}/6.04\ \text{in}^2 = 5.96\ \text{ksi}$$

$$f_c/F_{cy} = 5.96/38 = \underline{.157}$$

The Retainer is secured to the rotor arms by six rows of 5 screws each. The screws are ¼ dia., aircraft quality flat-head screws. It is assumed that each screw carries 1/30th of the load. The load is

$$18\ \text{lb bomb} \times 2000\ g = 36\ \text{k}$$

$$2\tfrac{1}{2}\ \text{lb retainer} \times 2000\ g = \underline{\ \ 5\ \text{k}}$$

$$41\ \text{k}$$

The bearing stress of the screws on the retainer is

$$f_{br} = 41\ \text{k}/(30 \times \tfrac{1}{4}'' \times \tfrac{1}{4}'') = 21.9\ \text{ksi}$$

$$f_{br}/F_{bry} = 21.9/74 = \underline{.296}$$

The average shear stress in the retainer material adjacent to the bolt lines is $f_s = S/A$

where S = shear force = 41 k/8

A = shear area = ¼ in x 4 in = 1 in^2

$$f_s = 41\ \text{k}/8 \times 1\ \text{in}^2 = 5.12\ \text{ksi}$$

$$f_s/F_{su} = 5.12/38 = \underline{.135}$$

Combining the shear and compressive stress ratios gives

$$f_c/F_{cy} + f_s/F_{su} = .157 + .135 = \underline{.292}$$

The shear stress in the ¼" dia screws is

$$f_s = 41\ \text{k}/(30 \times .0491\ \text{in}^2) \quad 27.9\ \text{ksi}$$

$$f_s/F_{su} = 27.9/75 = \underline{.372}$$

<u>Bomb Retainer - Arm Attachment DWG 1162</u>

Four of the six lines of screws fasten directly to the arms.

The depth of the holes in the arms is 9/16 in. Hence the bearing stress of the screws on the arm is not critical.

The remaining two lines of screws fasten to 3/16" thick 2024-T4 plates (P/N 1162-36). The bearing stress ratio is

$$f_{br}/F_{bry} = .296 \times (.25/.1875) = \underline{.395}$$

The average shear stress ratio in the plate material adjacent to the bolt line is

$$f_s/F_{su} = .135 \times (.25/\ .1875) = \underline{.180}$$

The load is transferred from the plates to the arms by six AN4 bolts per plate. Since the load is put into the plates by five ¼" dia screws, the bolt joints are not critical.

<u>Rotor Arms DWG 1162</u>

Material: 2024-T351 3/4" thick aluminum plate

<u>Cross-sectional area of arm in region of retainer.</u>

$$A = 5\tfrac{1}{2}'' \times 3/4'' - A_1 - A_2 - A_3 - A_4$$

where A_1 = Area removed to fit 5½" dia retainer

A_2 = Area removed by 1½" slot in side

A_3 = Area removed for screws and nuts

A_4 = Area removed due to 10° end taper top and bottom

$$A_1 = \pi(5.5)^2\ 70^{\circ}/(360^{\circ} \times 4) - (5.5)^2(\mathrm{Cos}\ 35^{\circ})(\mathrm{Sin} 35^{\circ})/4$$

$$= 4.62\ \mathrm{in}^2 - 3.56\ \mathrm{in}^2 = 1.060\ \mathrm{in}^2$$

$$A_2 = .30'' \times 1.5'' = .450\ \mathrm{in}^2$$

$$A_3 = 2 \times .25'' \times .5625'' + .3125'' \times .5625''$$

$$= .281 + .176 = .457\ \mathrm{in}^2$$

$$A_1 + A_2 + A_3 = 1.967\ \mathrm{in}^2$$

At the screw, closest to the center of rotation (first screw)

$$A_4 = 2 \times .75'' \times .75''\ (\mathrm{Tan}\ 10^{\circ}) = .198\ \mathrm{in}^2$$

$$A = 4.125 - 1.967 - .198 = 1.960 \text{ in}^2$$

At the screw second closest to the center of rotation,

$$A_4 = 2 \times .75'' \times 1.5'' \ (\text{Tan } 10^\circ) = .396 \text{ in}^2$$

$$A = 4.125 - 1.967 - .396 = 1.762 \text{ in}^2$$

critical region is at the first screw.

At the end of the arm

$$A_4 = 2 \times .75 \times 4.5 \ (\text{Tan } 10^\circ) = 1.190 \text{ in}^2$$

$$A = 4.125 - 1.967 - 1.190 = .968 \text{ in}^2$$

The average area between the first screw and the end is

$$\bar{A} = (1.960 - .968)/2 = 1.465 \text{ in}^2$$

The weight of the material is

$$w = 1.465 \text{ in}^2 \times 3.75 \text{ in}^2 \times .1 \text{ lb/in}^3 = .549 \text{ lb}$$

The load at the first screw is

$$41 \text{ k}/3 + 2000 \times .549 \text{ lb} = 13.7 + 1.1 = 14.8 \text{ kips}$$

$$f_t = 14.8 \text{ k}/ \ 1.960 \text{ in}^2 = 7.5 \text{ ksi}$$

$$f_t/F_{ty} = 7.5/ \ 40 = \underline{.188}$$

<u>Cross-sectional area of arm in region of plate attachment (P/N 1162-36)</u>.

$$A = 5.5 \text{ in} \times .75 \text{ in} - A_1 - A_2 - A_3$$

where A_1 = area removed to fit retainer = 1.060 in^2

A_2 = area removed by slot in side

A_3 = area removed for AN4 bolts.

$$A_2 = .35 \text{ in} \times 2.2 \text{ in} = .77 \text{ in}^2$$

$$A_3 = .25 \text{ in} \times .9375 \text{ in} = .234 \text{ in}^2$$

$$A = 4.125 - 1.060 - .77 - .234 = 2.06 \text{ in}^2$$

The weight of the arm material outboard of this point is not greater than

$$w = 2 \text{ in}^2 \times 7 \text{ in} \times .1 \text{ lb/in}^3 = 1.4 \text{ lb}$$

The load is

$$41 \text{ k}/2 + 2000 \times 1.4 = 23.3 \text{ kips}$$

$$f_t = 23.3 \text{ k}/ 2.06 \text{ in}^2 = 11.3 \text{ ksi}$$

$$f_t/F_{ty} = 11.3/ 40 = \underline{.283}$$

General expression for stress in rotor arms.

Consider a thin rod with length b rotating about the point 0 and having a concentrated mass m_i at radius r_i

let f_{ta} = stress at a

A = cross sectional area

A_a = cross-sectional area at a

dm = elemental mass

ρ = density of rod material

ω = angular velocity

then $$(Af_t)_a = \int_a^b dm\,r\omega^2 + \sum_i m_i r_i \omega^2$$

substituting

$$dm = \rho A \, dr,$$

$$(Af_t)_a = \int_a^b \rho A\omega^2 r dr + \sum_i m_i r_i \omega^2$$

for ρ and A constant,

$$(Af_t)_a = \rho A\omega^2 (b^2 - a^2)/2 + \sum_i m_i r_i \omega^2$$

or

$$(Af_t)_a = \rho A\omega^2 b^2 (1 - a^2/b^2)/2 + \sum m_i r_i \omega^2 \quad (1)$$

Let station a be 19" from the center of the rotor toward the bomb, i.e., just inboard of the bomb. Then,

$a = 1.582$ ft

$b = 3$ ft

$\omega = 152$ rad/sec (1450 RPM)

$\rho = .1$ lb/cu. in./g

$g = 32.2$ lbm ft/lb sec^2

$A_a = .75$ in x 1 in x 2 = 1.5 in^2

$A = 2$ in^2

$m_1 r_1 = 20.5$ lb x 33 in/(12 g)

$\omega^2/g = 745$ g/ft

$(Af_t)_a = .1x2x36x3x745x(1-19^2/36^2)/2+(20.5/2)x(33/12)x745$

$= 5810 + 21,000 = 26.81$ kips

$f_{ta} = 26.81$ k/1.5 = 17.9 ksi

$f_{ta}/F_{ty} = 17.9/\ 40 = \underline{.447}$

Let station a be 7.5 in from the center of the rotor, i.e., at the point of minimum area in the vicinity of the surge tanks.

$a = 7.5$ in

$Aa = 1.15$ in x 1.125 in x 2 = 2.59 in^2

$A = 2$ in^2

$m_2 r_2 = 36$ lb x 7.5 in/2g

m_3 = extra mass of aluminum in excess of A just outboard of the surge tank

$= 2.5 \text{ in} \times 10 \text{ in} \times 1.125 \text{ in} \times .1 \text{ lb/in}^3$

$= 2.81 \text{ lb}$

$r_3 = 12.5 \text{ in}$

$$(Af_t)_a = 8050(1-7.5^2/36^2) + 21{,}000 \text{ lb} + (18\times7.5 + 2.81\times12.5)\times745/12$$

$$= 7{,}760 + 21{,}000 + 10{,}550 \text{ lb} = 39.31 \text{ kips}$$

$$f_t = 39.31 \text{ k}/2.59 \text{ in}^2 = 15.2 \text{ ksi}$$

$$f_t/F_{ty} = 15.2/\ 40 = \underline{.38}$$

It is assumed above that the side plates (P/N 1162-4 and 1162-22) carry their share of the load, i.e. approximately 1/3 of the 39.31 kips above must be transferred to the side plates via six AN6 bolts. The bolt area in shear is

$$A = 6 \times .1105 \text{ in}^2 = .663 \text{ in}^2$$

$$f_s = 13.1 \text{ k}/\ .663 \text{ in}^2 = 19.75 \text{ ksi}$$

$$f_s/F_{su} = 19.75/75 = \underline{.264}$$

The bearing stress of the bolts on the side plate is

$$f_{br} = 13.1 \text{ k}/(6 \times .375 \times .375) = 15.5 \text{ ksi}$$

$$f_{br}/F_{bru} = 15.5/95 = \underline{.164}$$

The maximum stress of the 1½" dia pin (DWG 1158B) on the rotor arms is

$$f_{br} = 36\text{k}/(1.5 \text{ in} \times 1.125 \text{ in}) = 21.3 \text{ ksi}$$

$$f_{br}/F_{bry} = 21.3/74 = \underline{.289}$$

The portion of the rotor between the center and the

counter-weight can be analyzed in a manner similar to the preceding. However, DWG 1162 sheet 1, shows the foregoing simplifying assumptions regarding cross-sectional area to be conservative for the counter-weight half of the rotor. Hence similar calculations are not required.

Counter-weight Attachment DWG 1162

The load to be carried in double shear by 7 AN6 bolts is 41 kips.

$$f_s = 41\ \text{k}/(7 \times .1105\ \text{in}^2 \times 2) = 25.5\ \text{ksi}$$

$$f_s/F_{su} = 25.5/75 = \underline{.354}$$

The bearing stress on the arms is

$$f_{br} = 41\ \text{k}/(7 \times .375 \times .75 \times 2) = 10.4\ \text{ksi}$$

$$f_{br}/F_{bry} = 10.4/74 = \underline{.151}$$

The minimum arm cross-section in this region is

$$A = 2 \times .75\ \text{in} \times (2\ \text{in} - .375\ \text{in}) = 2.44\ \text{in}^2$$

$$f_t = 41\ \text{k}/2.44\ \text{in}^2 = 16.8\ \text{ksi}$$

$$f_t/F_{ty} = 16.8/40 = \underline{.42}$$

Charging Valve Attachment P/N 1162-14

Valve weight	.13 lb
Cap weight	.185 lb
Total	.315 lb

Assume that the coupler fitting (P/N 1162-12) supports the valve and cap.

Radius for bending = 3", bending moment arm = 3/4"

Radius for tension = 4.5"

Bending moment = .315 lb x (3"/12") x 745 x 3/4"= 44 in-lb

Tensile force = .315 lb x (4.5 in/12) x 745 = 87.9 lb

The tension in the inboard pair of #8-32 steel screws is

87.9 lb/2 + 44 in-lb/.83 in = 97 lb

f_t = 97 lb/(2 x .0139 in^2) = 3.49 ksi

f_t/F_{ty} = 3.49/35 = <u>.10</u>

The shear stress in the aluminum thread is

f_s = 97 lb/(π x .1437 x .375) = 574 psi

f_s/F_{su} = .574/38 = <u>.015</u>

<u>Bomb Assembly DWGS 1131B, 1132A</u>

Material: 321 Stainless Steel (annealed)

The Bomb is designed in accordance with the ASME Unfired Pressure Vessel Code (Ref.5) for a working pressure of 3000 psig. It was hydrostatically tested to 4500 psig on 17 August 1965.

<u>Cylinder</u>.

Hoop stress:

From Ref. 5, $t = PR/(SE - 0.6\ P)$

where

t = required wall thickness, inches

P = working pressure (3000 psi)

S = allowable stress (18750 psi -- Ref. 5)

E = efficiency of longitudinal joint (=1)

R = inside radius of cylinder (2")

t = (3000 psig x 2)/(18750 psig - 0.6 x 3000 psig)= .354 in.

Longitudinal stress:

From Ref. 5, $t = PR/(2SE - 0.4P)$

where $E = 0.9$ for fully X-rayed joint (Ref. 5)

= (3000 psig x 2)/(2 x 18750 x 0.9-0.4 x 3000 psig)

= .1843 inches.

Hence a cylinder wall 3/8 inch thick is adequate.

Hemispherical cap.

From Ref. 5, $t = PR/(2SE - 0.2P)$

t = (3000 psig x 2 in)/(2 x 18750 psig x 0.9-0.2 x 3000 psig)

= .182 in

Hence a wall thickness of 1/4 inch is adequate.

Closure.

The Closure is in the form of a reducer section as recommended by Ref. 5. The wall thickness is 3/8 inch except at the opening where it is a minimum of ¼ inch. The required thickness for hoop stress at the opening is:

$t = PR/ (SE - 0.6P)$

= 3000 psig x .9375 in/(18750 - 0.6 x 3000 psig)

= .166 in

Hence t = ¼ inch is adequate.

The bomb is sealed by an aluminum plug and an aluminum collar with a 2½-8 thread.

The force p on the plug is

$$p = \pi D^2P/4$$

where $D = 1.875''$

$$P = 3000 \text{ psig}$$

$$p = \pi(1.875^2) \times 3000 \text{ PSIG}/4$$

$$p = 8{,}320 \text{ lb}$$

The thread area in shear is

$$A = \pi \; DL/2$$

where D = pitch diameter = 2.347 in

L = thread length = ½ inch

$A = \pi \; 2.347 \times .5/2 = 1.845 \text{ in}^2$

$f_s = 8320 \text{ lb}/1.845 \text{ in}^2 = 4.5 \text{ ksi}$

The two ¼ NPTF openings in the closure are reinforced by bosses. The cross-sectional area removed by each opening is approximately ½" x 3/8" = 3/16 in^2. The cross-sectional area added by the 1¼ dia x ¼" high boss is (1¼ - ½)x ¼" = 3/16 in^2.

The Bomb is suspended in a retainer by a 4-15/16" - 8N Class 2A thread on the Closure. At 2000 g, the shear force is 36 kips. The thread shear area is

$$A = \pi \; DL/2$$

where D = pitch diameter = 4.86 in

L = length of thread = 1 in

$A = \pi \times 4.86 \text{ in} \times 1 \text{ in}/2 = 7.63 \text{ in}^2$

$f_s = 36 \text{ kips}/7.63 \text{ in}^2 = 4.72 \text{ ksi}$

Suspending the Bomb from the Closure causes an additional longitudinal stress in the cylinder due to the weight of the cylinder (lo lb) and the hemispherical cap (2 lb). At 2000 g, the tensile force is 24 kips.

$$f_{t1} = 24 \text{ kips} \times 4/\pi(D^2 - d^2)$$

where $D = 4\text{-}3/4$ in

$d = 4$ in

$$f_{t1} = 24k \times 4/\pi(22.7-16) = 4.57 \text{ ksi}$$

The longitudinal stress due to internal pressure is

$$f_{t2} = PR/2t$$

where $P = 3000$psig

$R = 2$ in

$t = 3/8$ in

$$f_{t2} = 3000 \text{ psig} \times 2 \text{ in}/(2 \times 3/8 \text{ in})$$

$$f_{t2} = 8 \text{ ksi}$$

$$f_t = f_{t1} + f_{t2} = 4.57 \text{ ksi} + 8 \text{ ksi} = 12.57 \text{ ksi}$$

The allowable longitudinal stress is

$$F_t = PR/2t$$

where $P = 3000$ psig

$R = 2$ in

$t = .1843$ in

$$F_t = 3000 \text{ psig} \times 2/(2 \times .1843 \text{ in}) = 16.3 \text{ ksi}$$

Hence, $f_t/F_t = 12.57/16.3 = \underline{.77}$

<u>Ballast Tank P/N 1162-49</u>

Material: 6061-T6 Aluminum

$F_{tu} = 42$ ksi $\quad F_{ty} = 35$ ksi

$F_{cy} = 34$ ksi $\quad F_{su} = 27$ ksi

$F_{bry} = 56$ ksi

Weight: 32-36 pounds

The ballast tanks are designed for a working pressure of 3000 psi and have been hydrostatically tested by the manufacturer to 5000 psi.

Hoop stress.

$$f_{tl} = Pr/t$$

where $P = 3000$ psi

$r = 3.25''$

$t = .525$ in

$$f_{tl} = 3000 \times 3.25\ \text{in}/.525\ \text{in} = 18.6\ \text{ksi}$$

$$f_{tl}/F_{ty} = 18.6/35 = \underline{.532}$$

Longitudinal stress.

Due to pressure

$$f_{tl} = Pr/2t = 18.6\ \text{ksi}/2 = 9.3\ \text{ksi}$$

Tension due to centrifugal force field from Equation (1)

$$f_{t2} = \rho\omega^2 b^2/2g$$

where $b = 14$ in $= 1.167$ ft

$\omega = 152$ rad/sec, $\rho = .1\ \text{lb/in}^3$

$\omega^2/g = 745$ lbf/lbmft

$$f_{t2} = .1 \times 745 \times 14 \times 1.167/2 = .608\ \text{ksi}$$

Bending stress due to centrifugal force field

Assume the tank to be a simply supported beam loaded by its own weight. At a radius of 7", the weight w of the tank is

$$w = 36\ \text{lb} \times 745 \times 7''/(12\ \text{in} \times 26.5\ \text{in}) = 590\ \text{lb/in.}$$

And the maximum bending moment (at a point of support) is

$$M = w\,x^2/2$$

where $w = 590$ lb/in

$x = 10.75$ in

$M = 590 \times (10.75^2)/2 = 34.05$ kip-in

The section modulus Z for the tank is

$$Z = \pi(D^4-d^4)/(32D)$$

where $D = 7.55$ in

$d = 6.5$ in

$$Z = \pi(7.55^4 - 6.5^4)/(32 \times 7.55) = 19.05 \text{ in}^3$$

$$f_b = M/Z = 34.05/19.05 = 1.79 \text{ ksi}$$

The total longitudinal stress is

pressure	9.3 ksi
tension	.608 ksi
bending	1.79 ksi
total	11.7 ksi

$f_t/F_{ty} = 11.7/35 = \underline{.334}$

Each ballast tank is held in place by a 4½ inch wide aluminum band and two band clamps. During rotation, a small off-center location of the tank c. g. would tend to make the tank slide farther out of position. However, with a static coefficient of friction of 1.4 (Ref. 3), the initial misalignment could be up to 7½ inches.

<u>Tubing</u>

Material: 321 stainless steel, $F_{tu} = 75$ ksi, $F_{ty} = 30$ ksi (minimum), 3/8 in O.C. x .058 in wall thickness

$$f_t = Pd/2t$$

where $P = 3$ ksi

$d = .259$ in

$t = .058$ in

$$f_t = 3 \times .259/(2 \times .058) = 6.7 \text{ ksi}$$

$$f_t/F_{ty} = 6.7/30 = \underline{.223}$$

$$f_t/F_{tu} = 6.7/75 = \underline{.0894}$$

¼" O. D. x .035 in wall thickness:

$d = .180$ in

$$f_t = 3 \times .180/(2 \times .035) = 7.72 \text{ ksi}$$

$$f_t/F_{ty} = 7.72/30 = \underline{.258}$$

$$f_t/F_{tu} = 7.72/75 = \underline{.103}$$

All tube fittings and valves are rated for a working pressure of at least 3000 PSI.

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1. Metallic Materials and Elements for Flight Vehicle Structures. MIL-HDBK-5, Dept of Defense, Wash. 25, D.C. Aug. 1962

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3. Mark's Mechanical Engineers Handbook, 6th Edition - Baumeister, Theodore (Editor) - McGraw Hill Book Co. 1958

4. Boston Gear Catalog 58, Boston Gear Works, Quincy 71, Massachusetts.

5. ASME Boiler and Pressure Vessel Code (1962), Section VIII and addenda, American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, N. Y. 10017

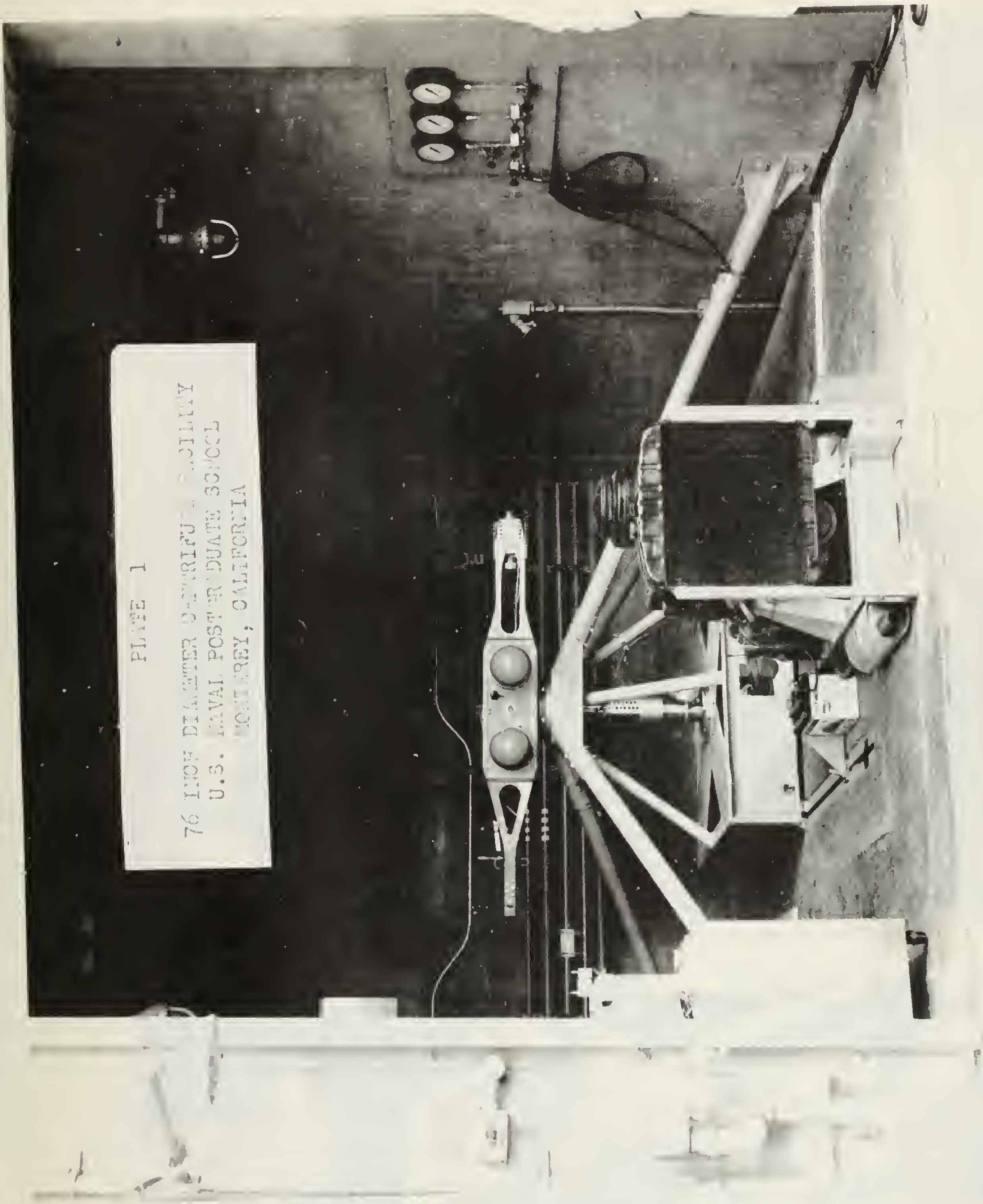


PLATE 1

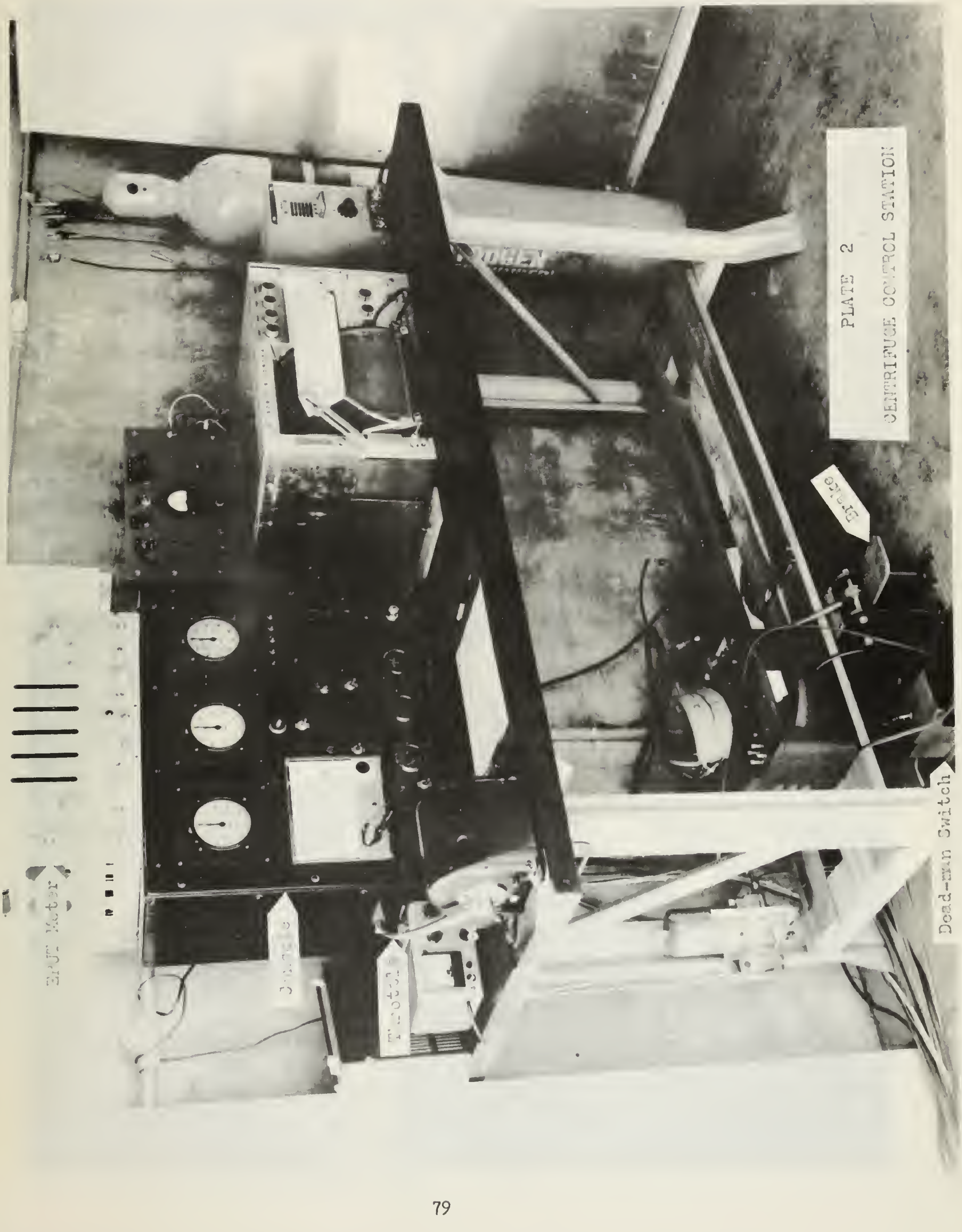


PLATE 2
CENTRIFUGE CONTROL STATION

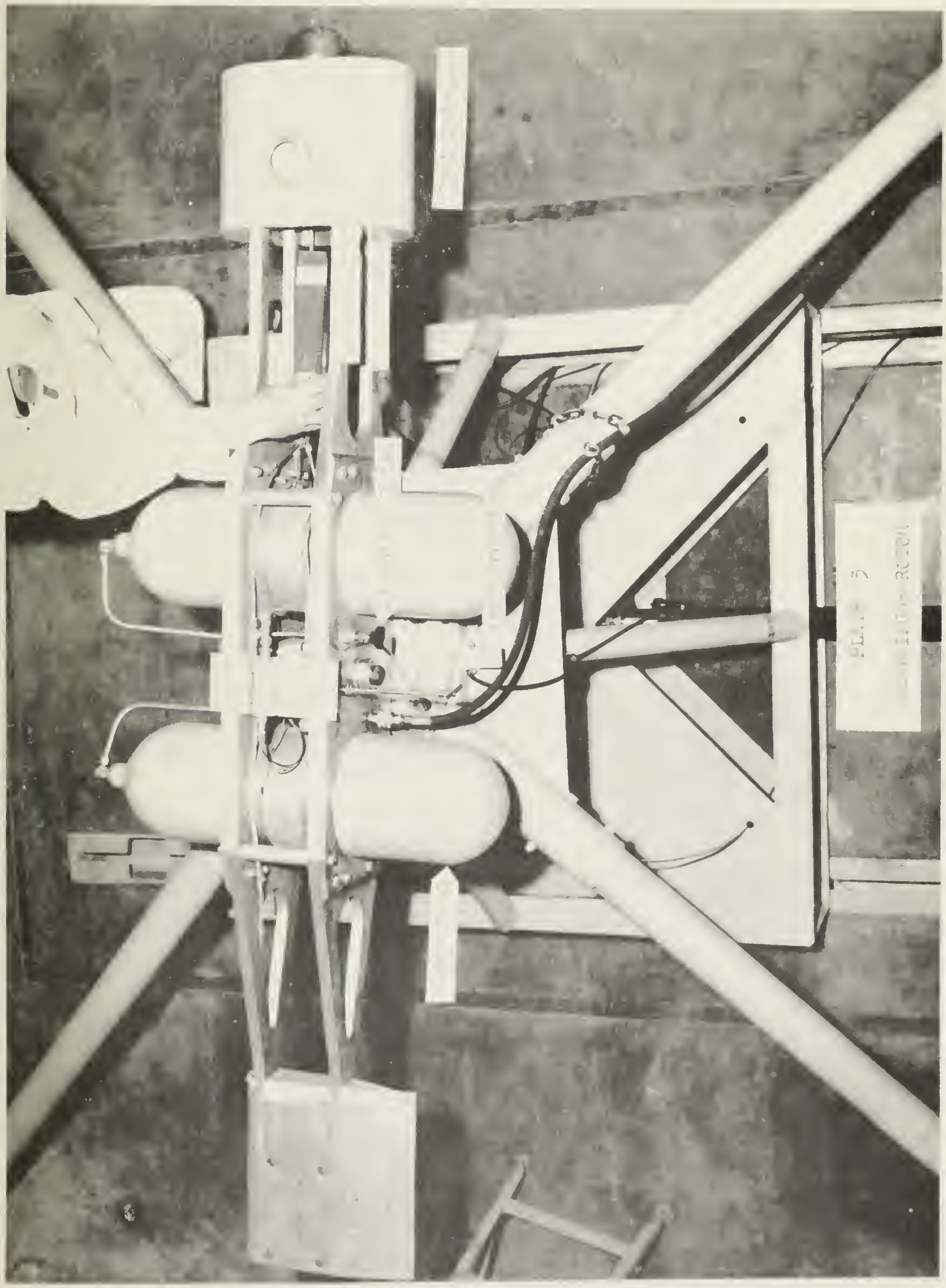


PLATE 5

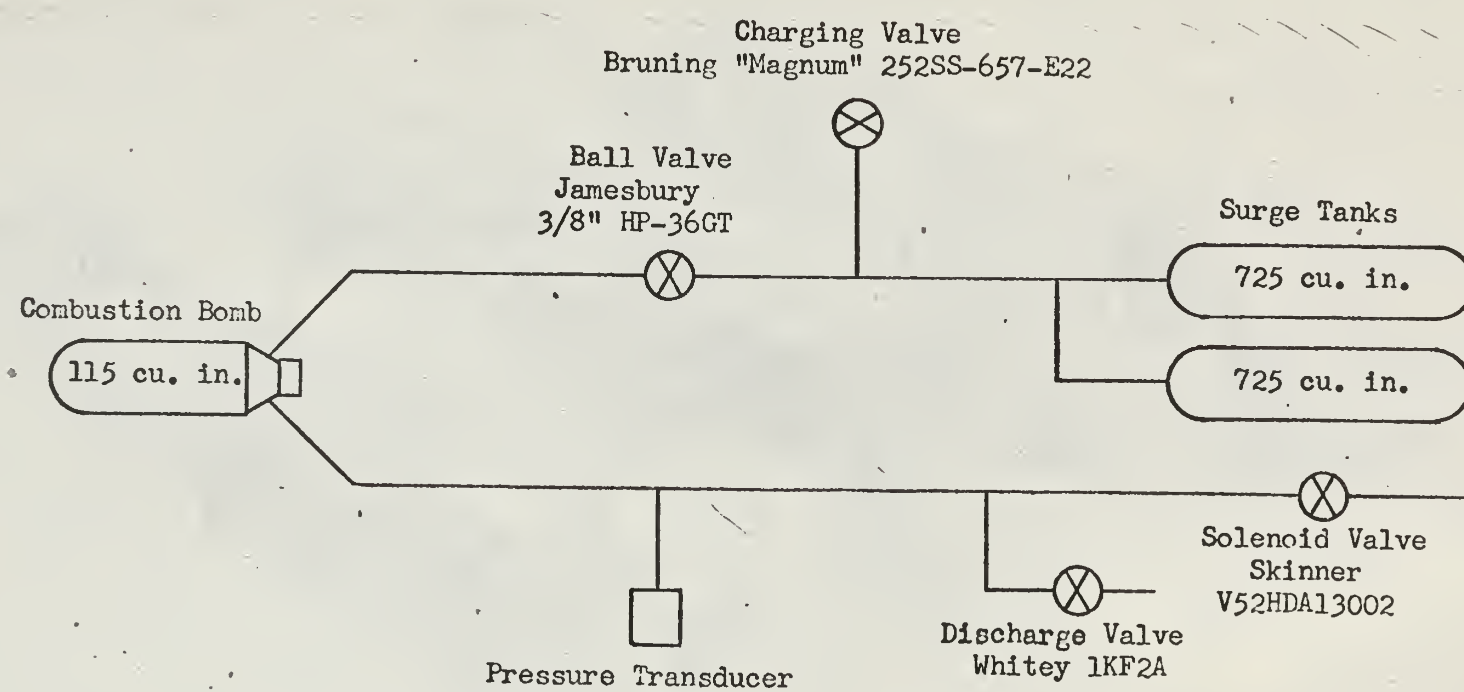


PLATE 4

COMBUSTION BOMB - SURGE TANK SYSTEM
(ROTOR SYSTEM)

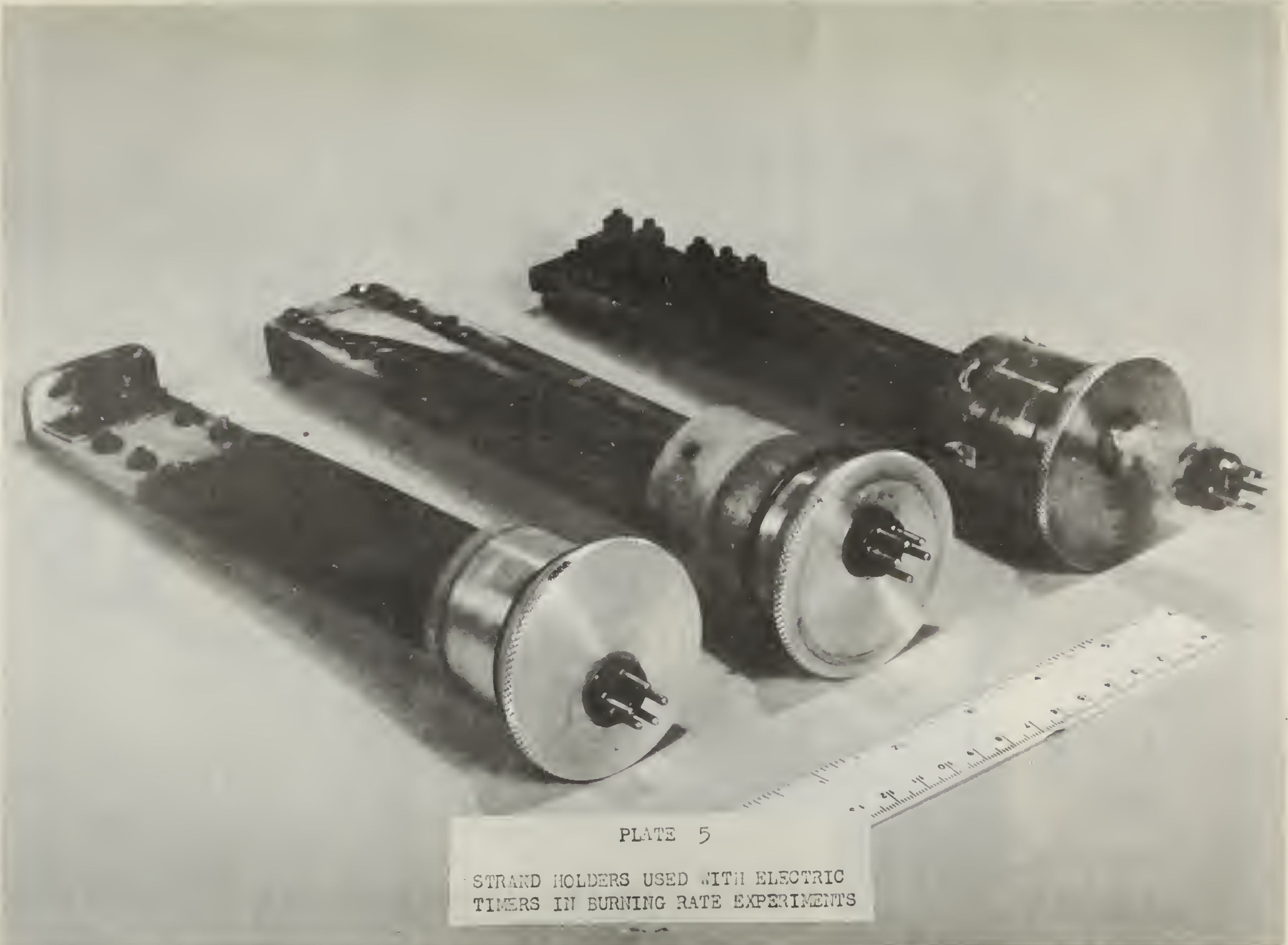


PLATE 5

STRAND HOLDERS USED WITH ELECTRIC TIMERS IN BURNING RATE EXPERIMENTS

PLATE 6

STRAND HOLDERS USED WITH PRESSURE INSTRUMENTATION IN BURNING RATE EXPERIMENTS

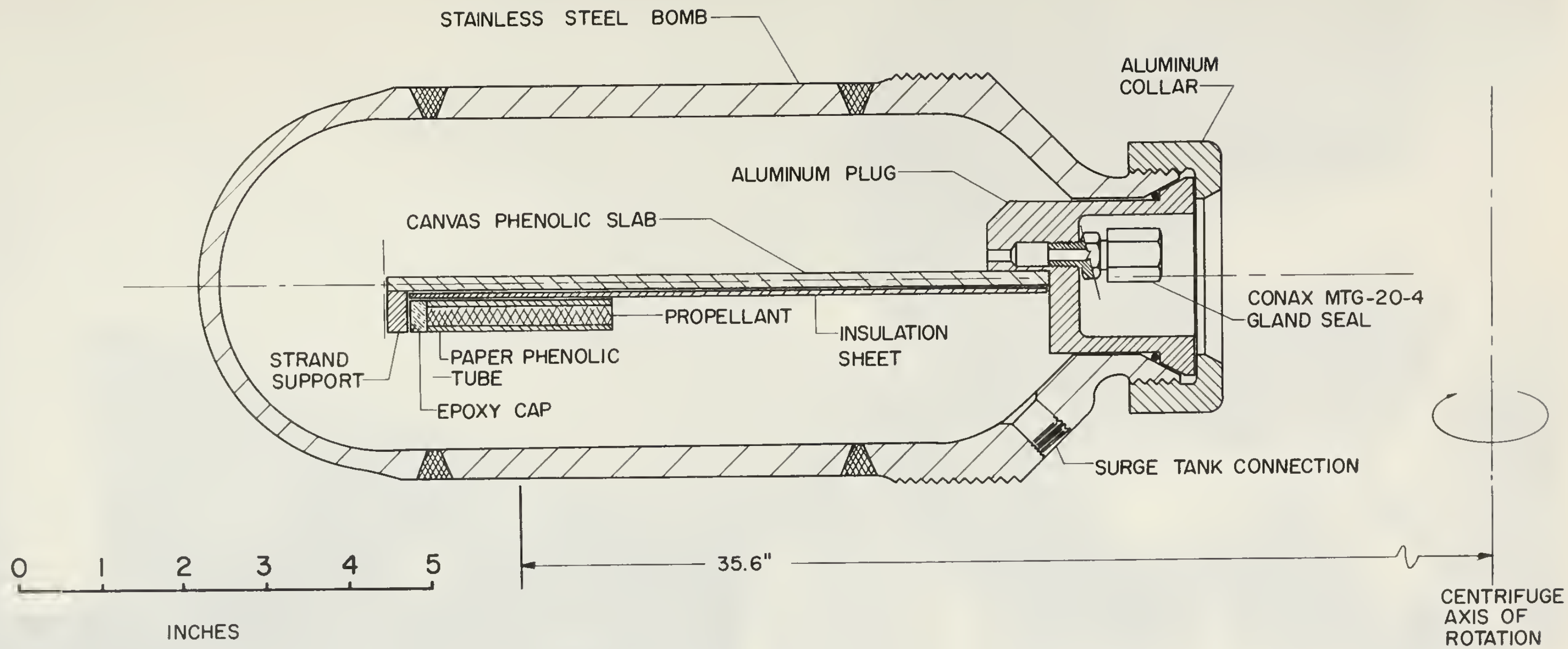


PLATE 7
STRAND HOLDER AND BOMB ASSEMBLY

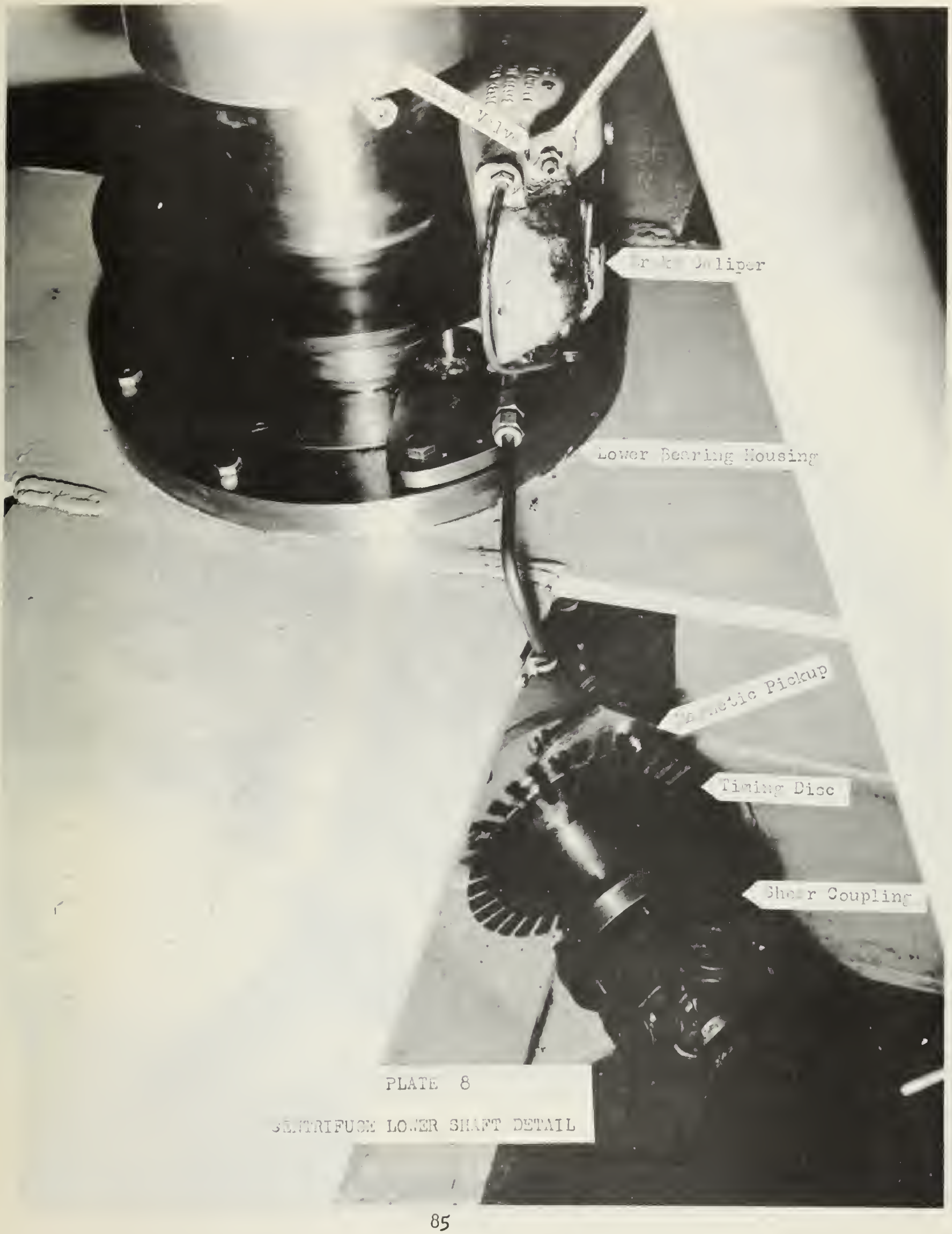


PLATE 8

CENTRIFUGE LOWER SHAFT DETAIL

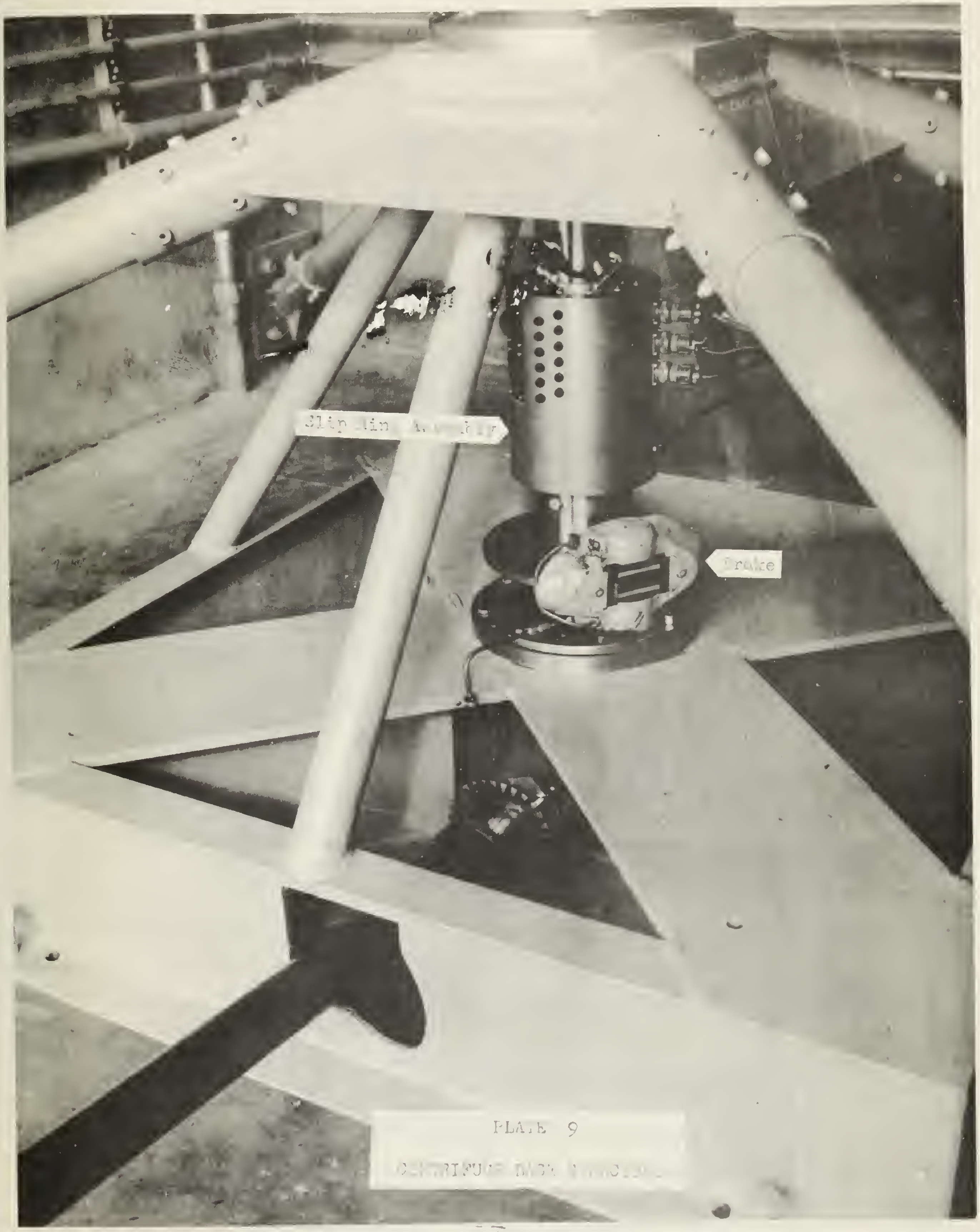


PLATE 9

CENTRIFUGE BASE [illegible]

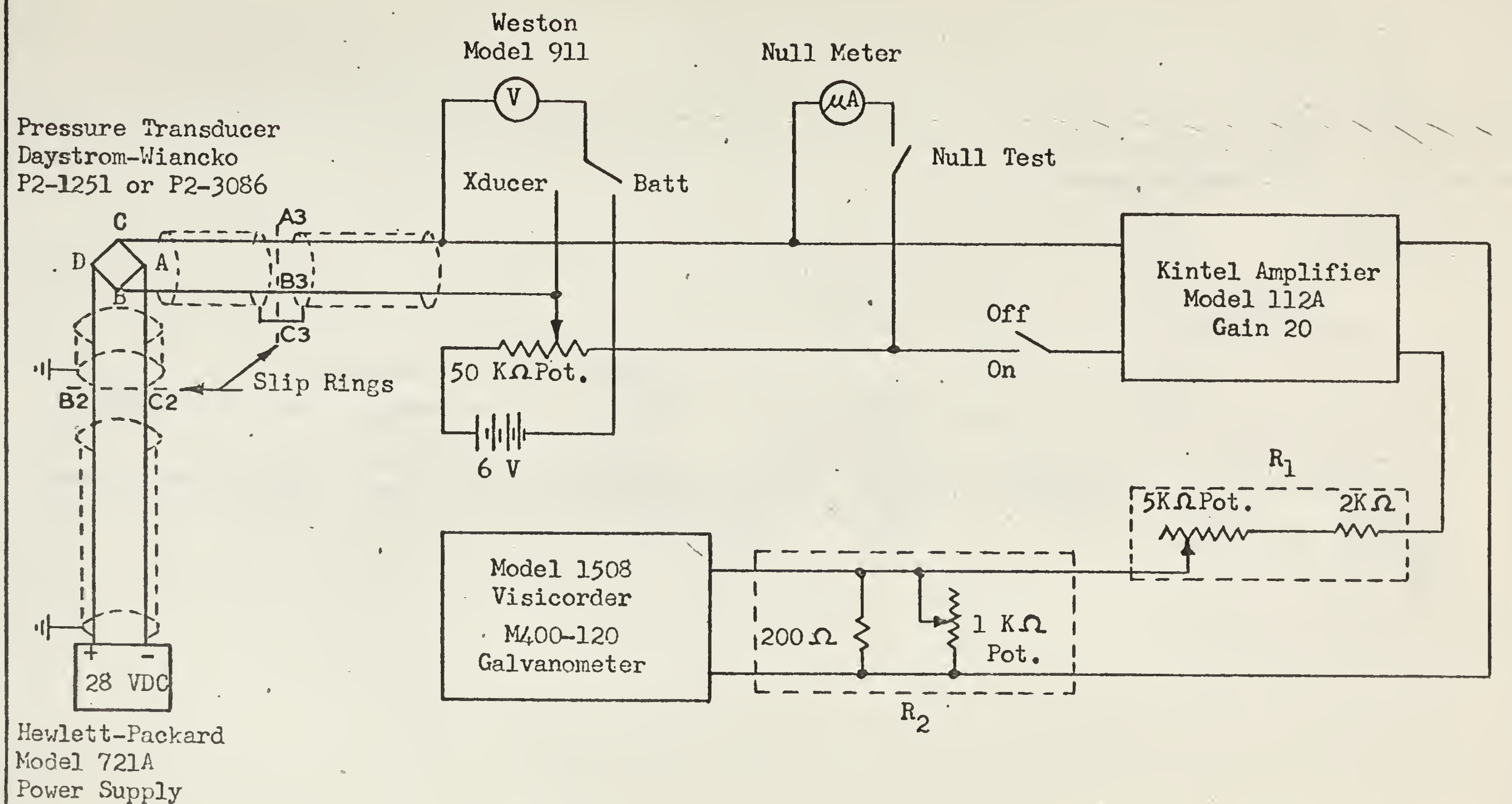


PLATE 10

PRESSURE CIRCUIT

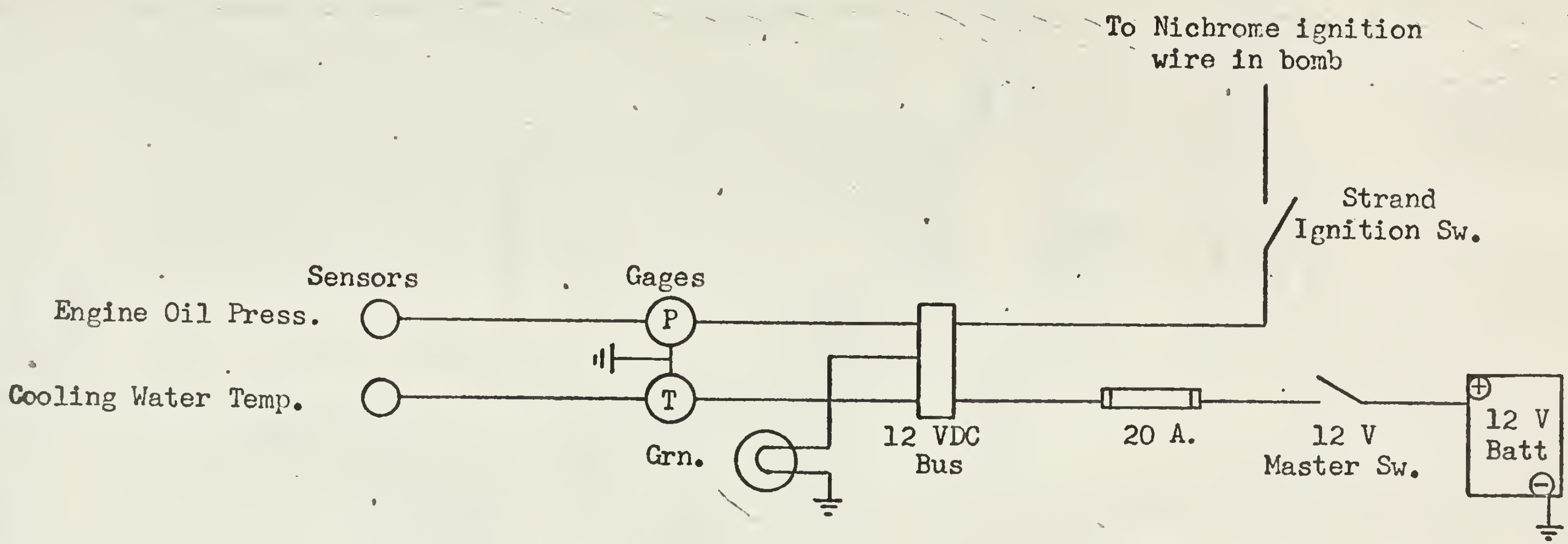


PLATE 11

12 VDC ELECTRICAL SYSTEM

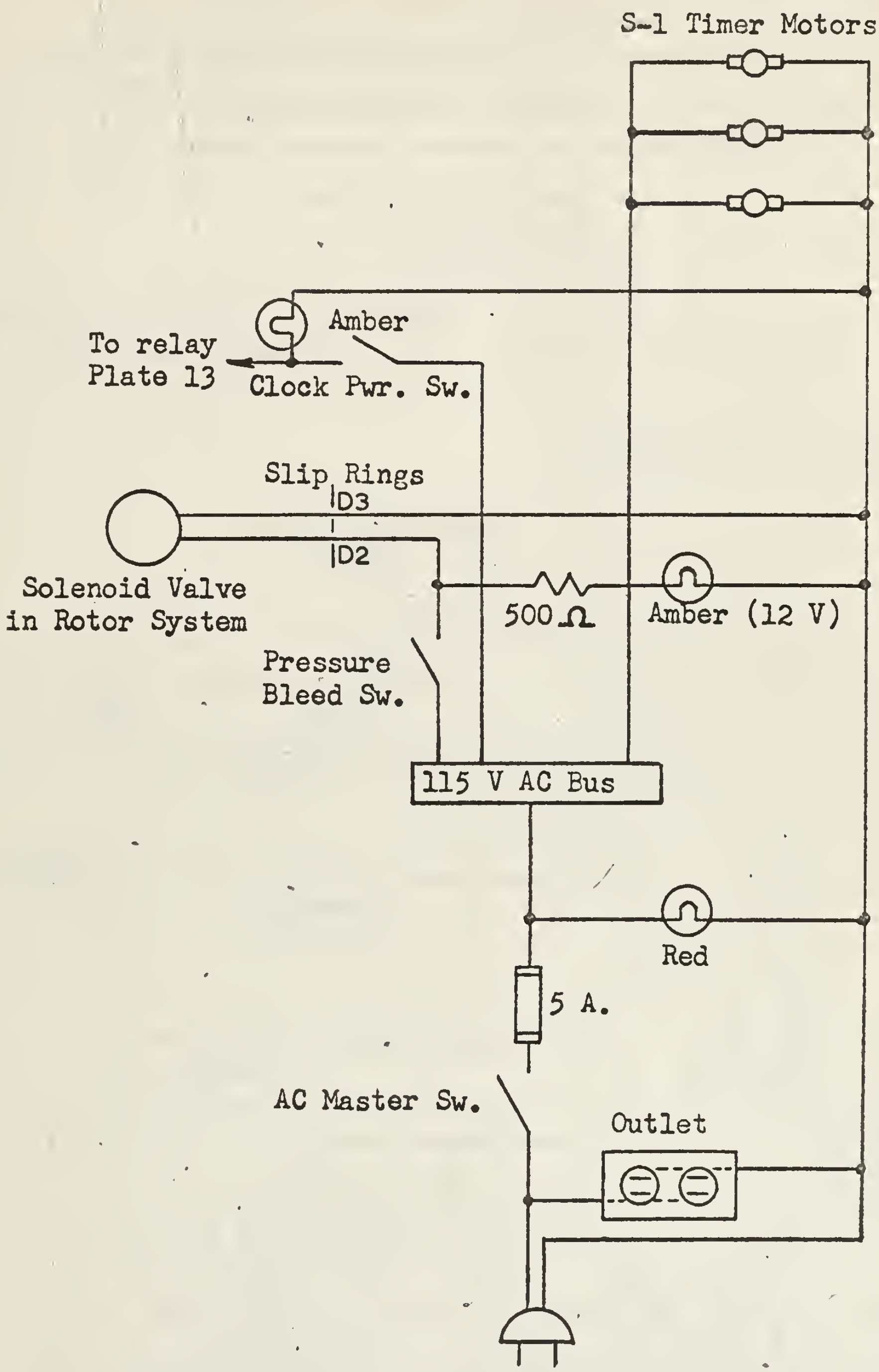


PLATE 12

115 V AC ELECTRICAL CIRCUIT

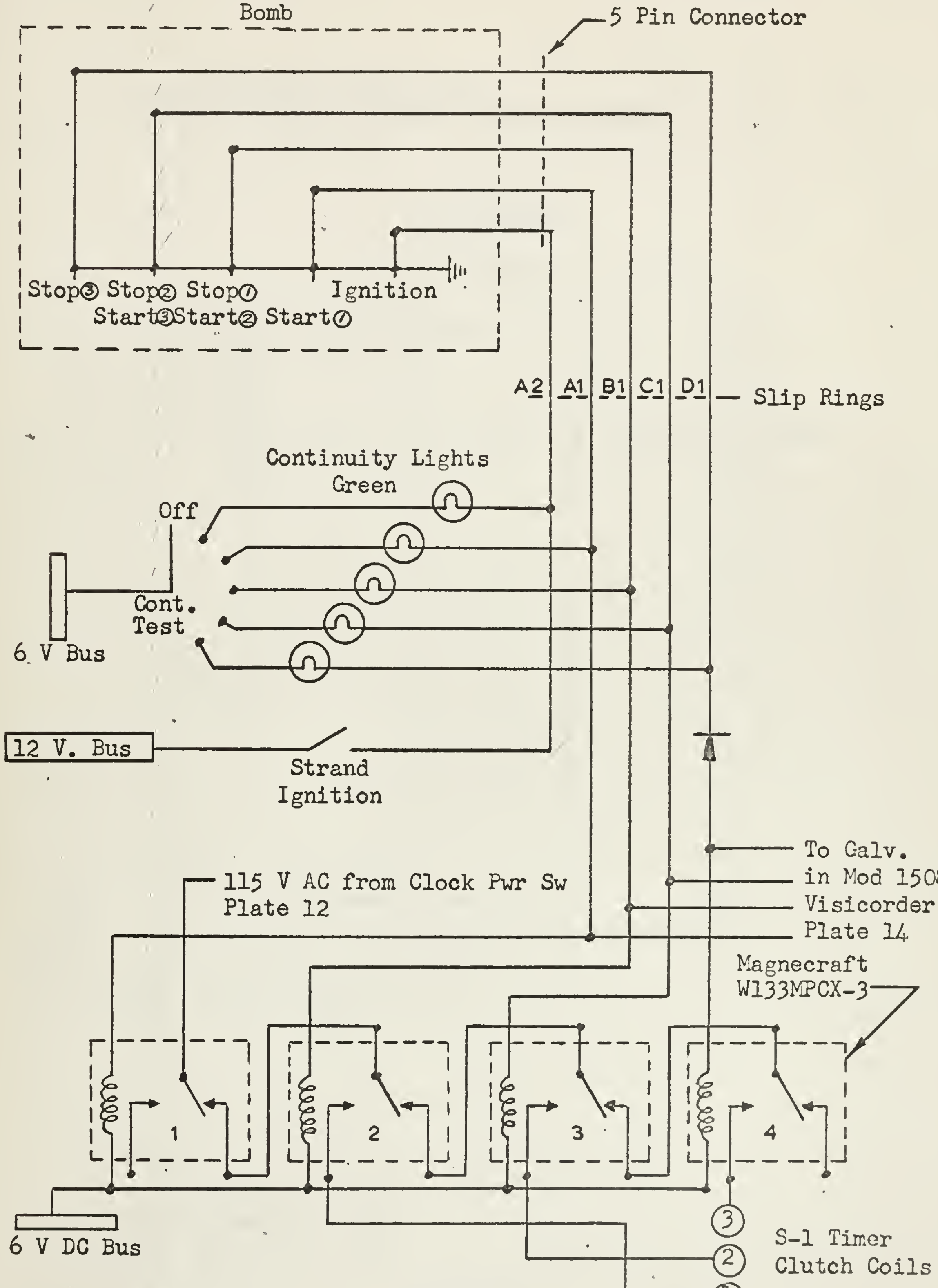


PLATE 13

TIMING, IGNITION, AND CONTINUITY-TEST CIRCUITS

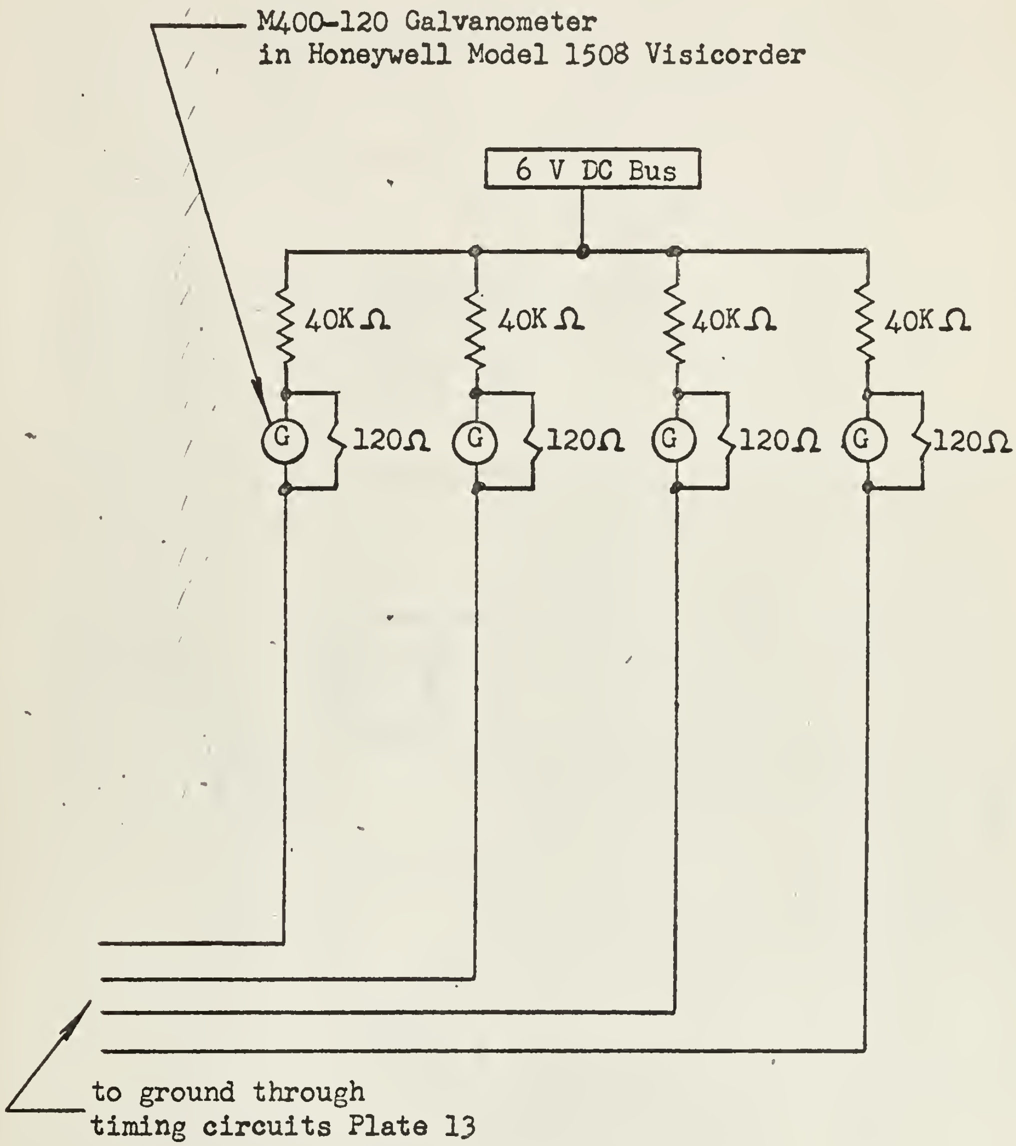


PLATE 14

VISICORDER TIMING CIRCUIT

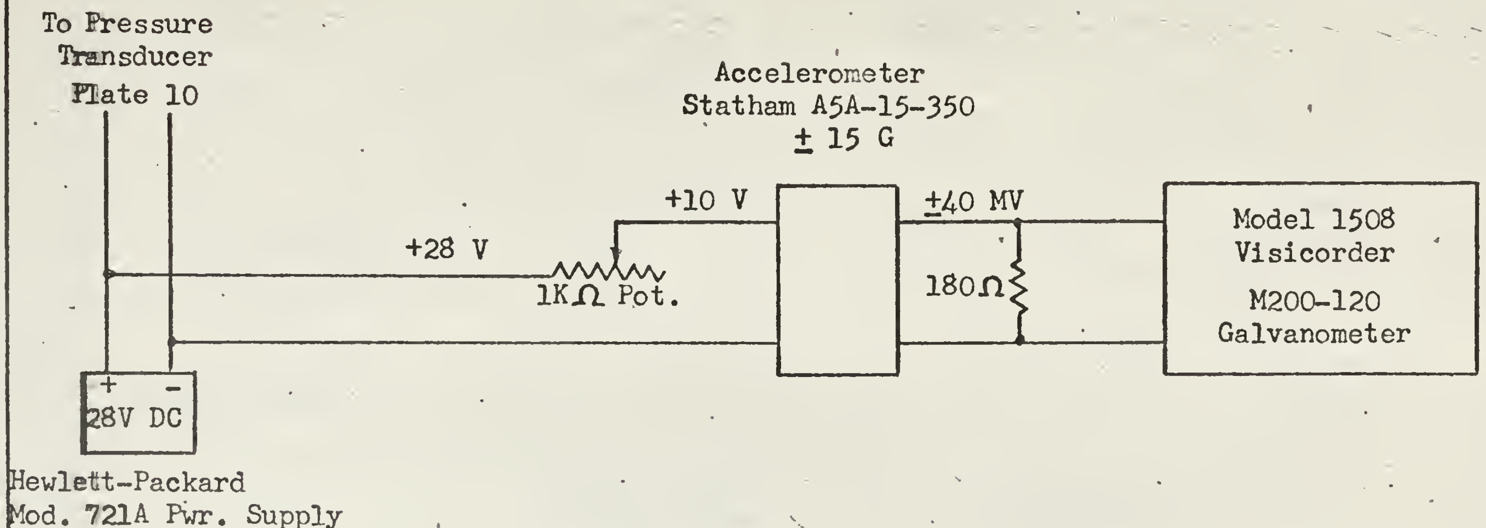


PLATE 15

VIBRATION PICK-UP CIRCUIT

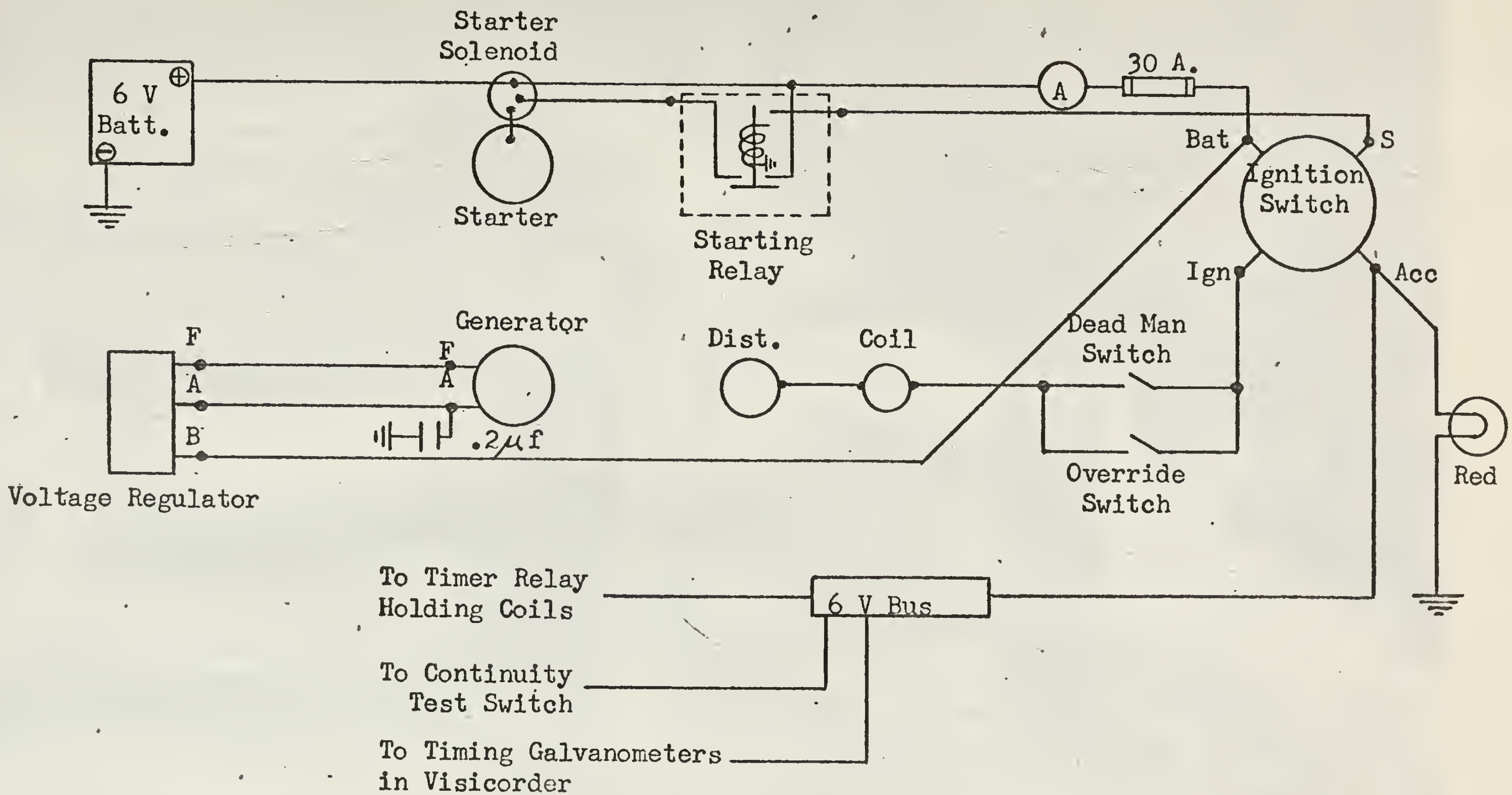


PLATE 16

6 V. DC ELECTRICAL SYSTEM

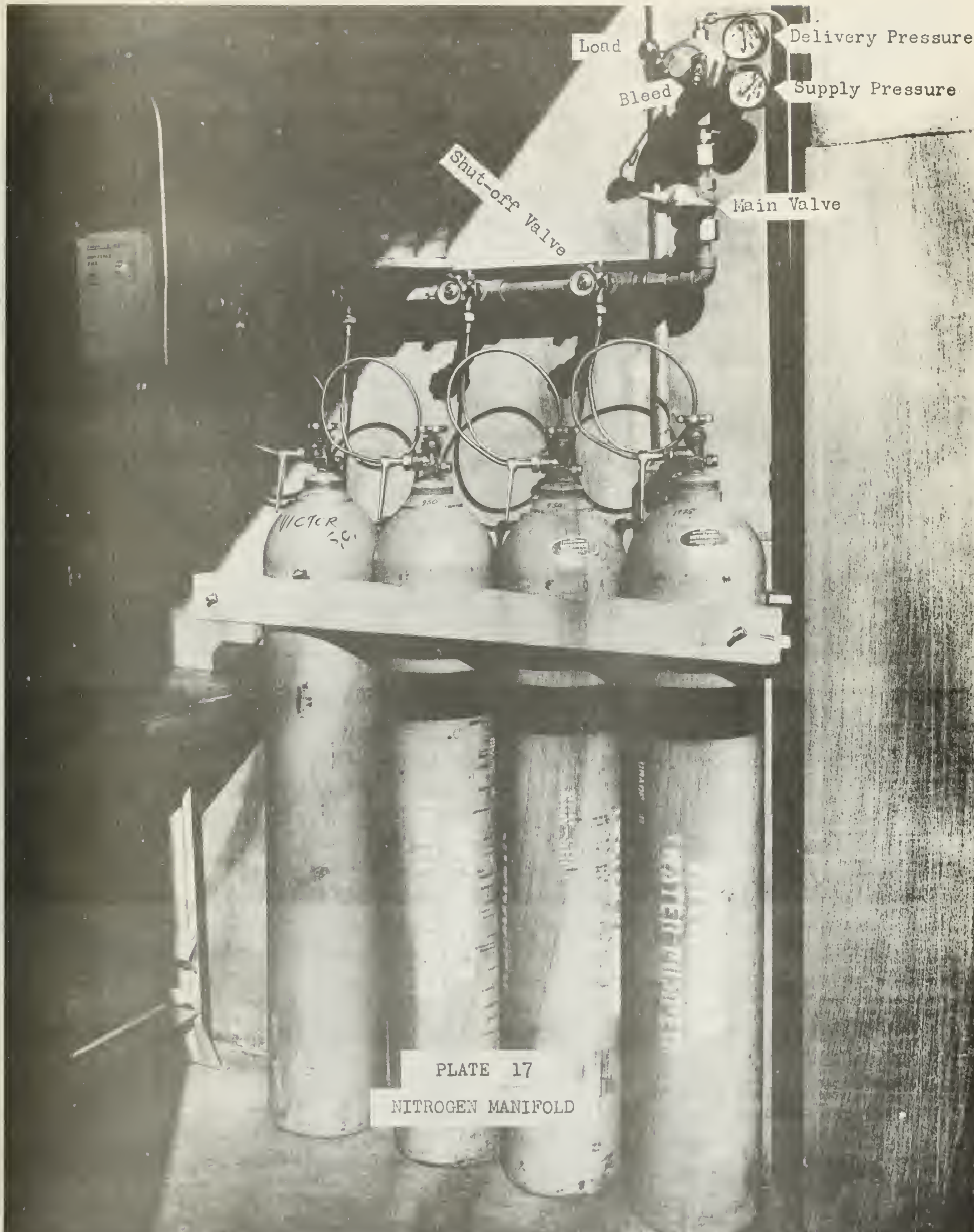


PLATE 17

NITROGEN MANIFOLD

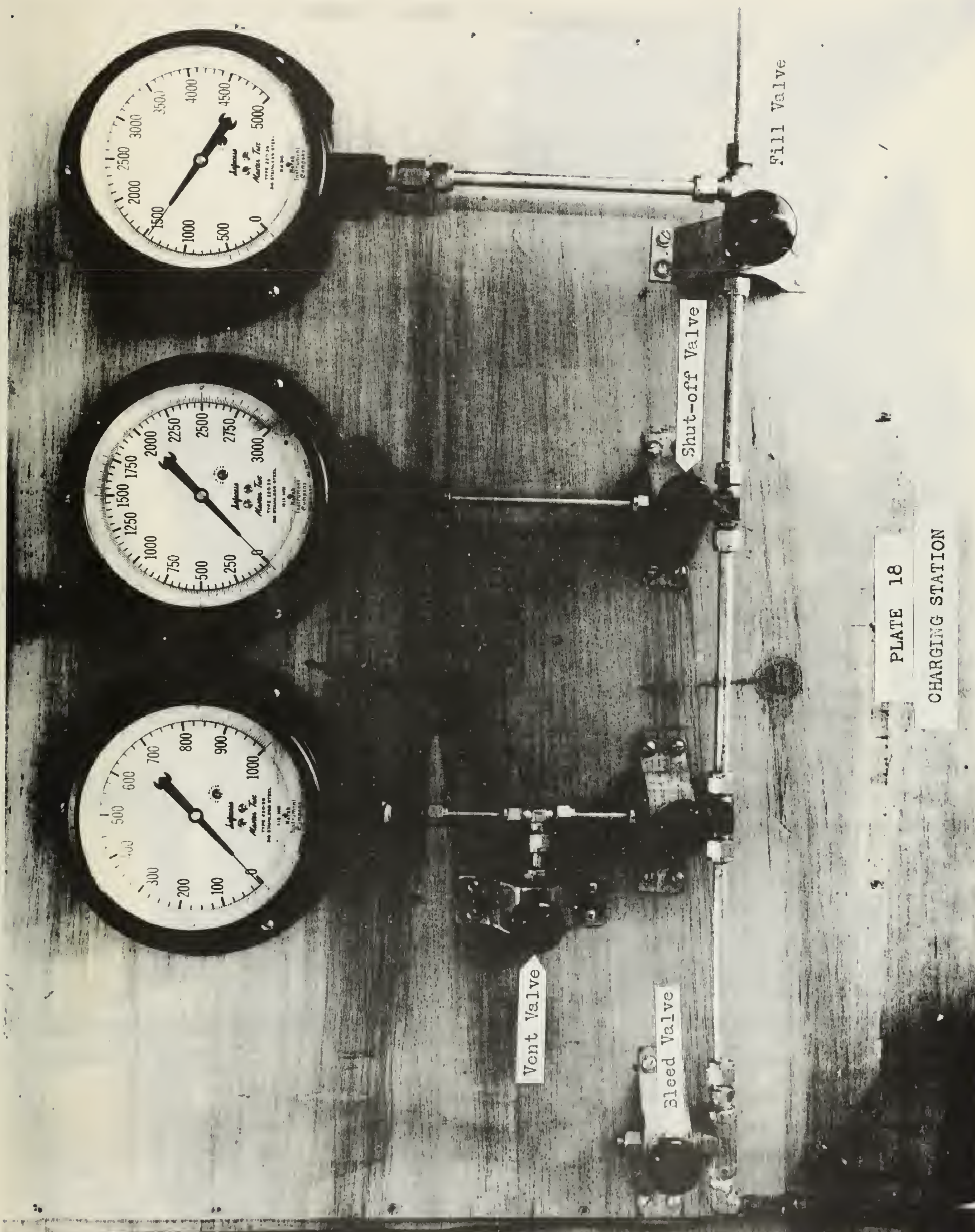


PLATE 18

CHARGING STATION

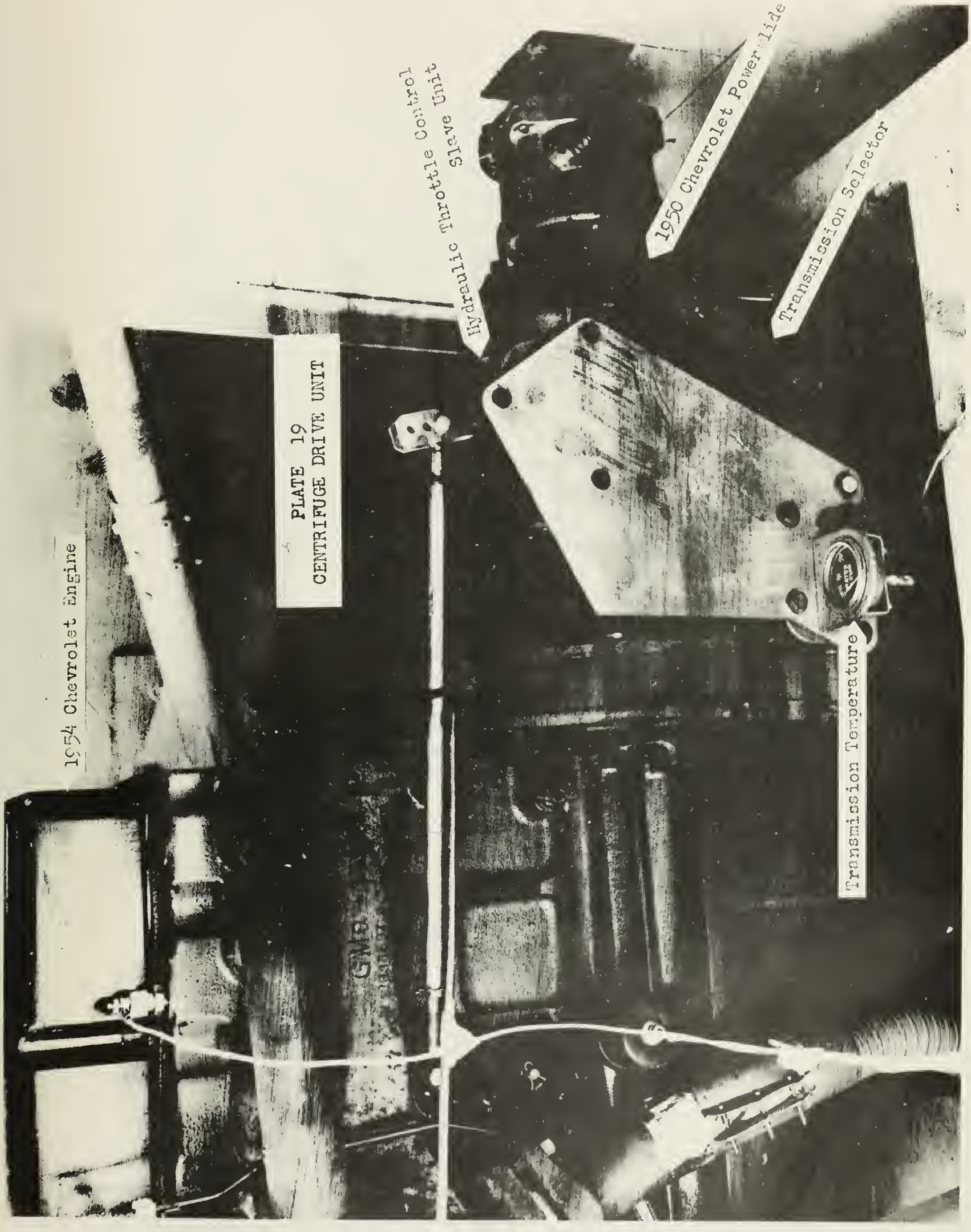


PLATE 19
CENTRIFUGE DRIVE UNIT

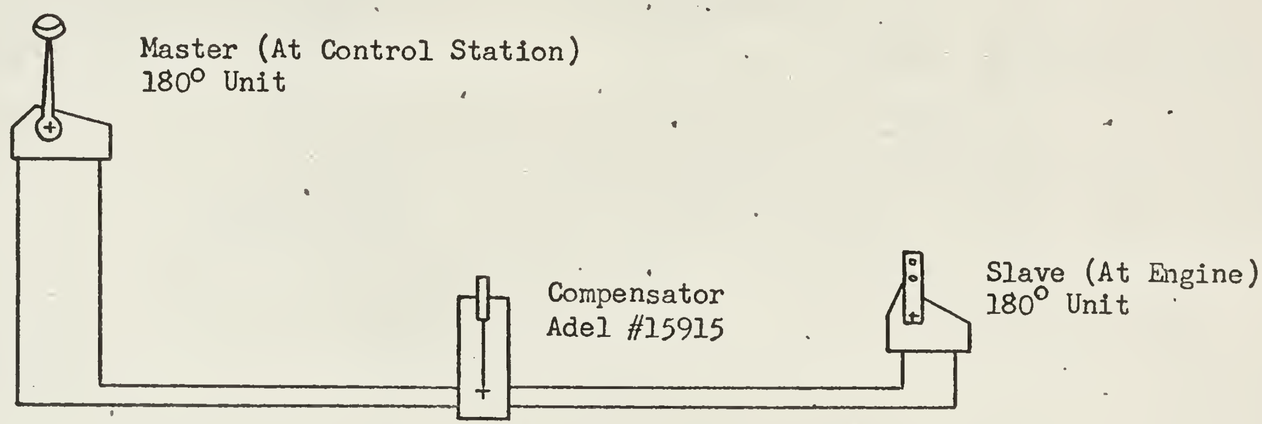


PLATE 20

CENTRIFUGE HYDRAULIC THROTTLE CONTROL

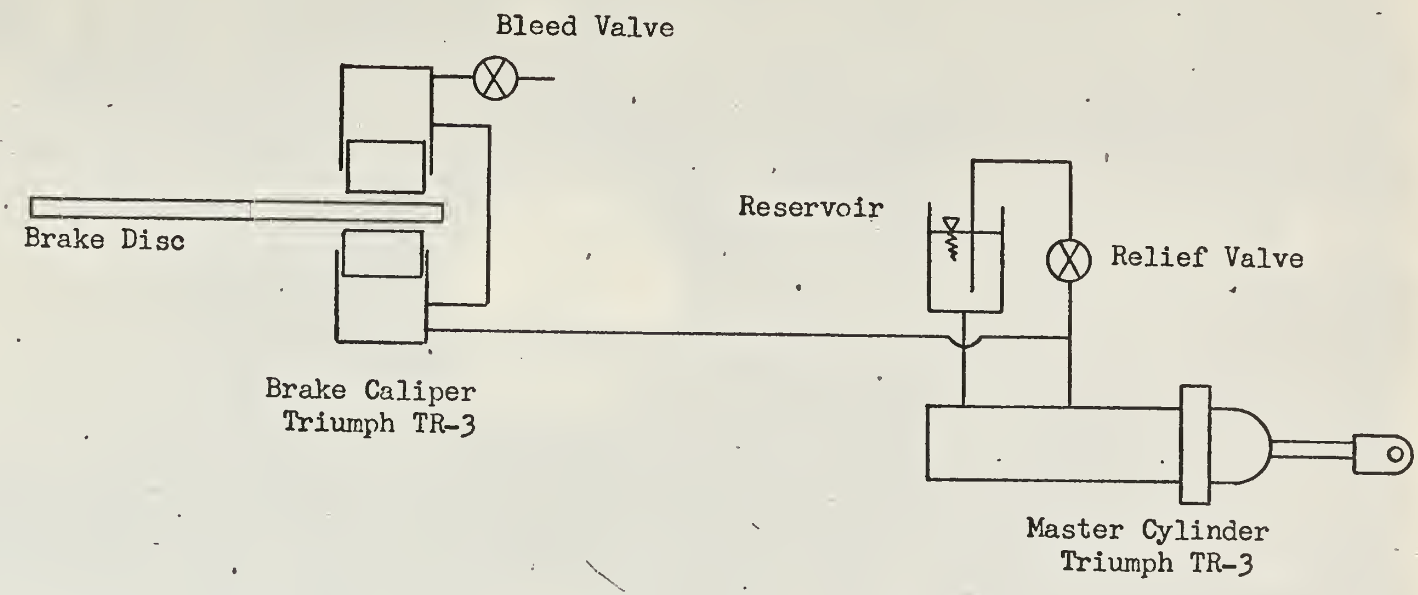


PLATE 21

CENTRIFUGE BRAKE SYSTEM

PLATF 22

VISICORDER SIGNAL CONDITIONING PANEL

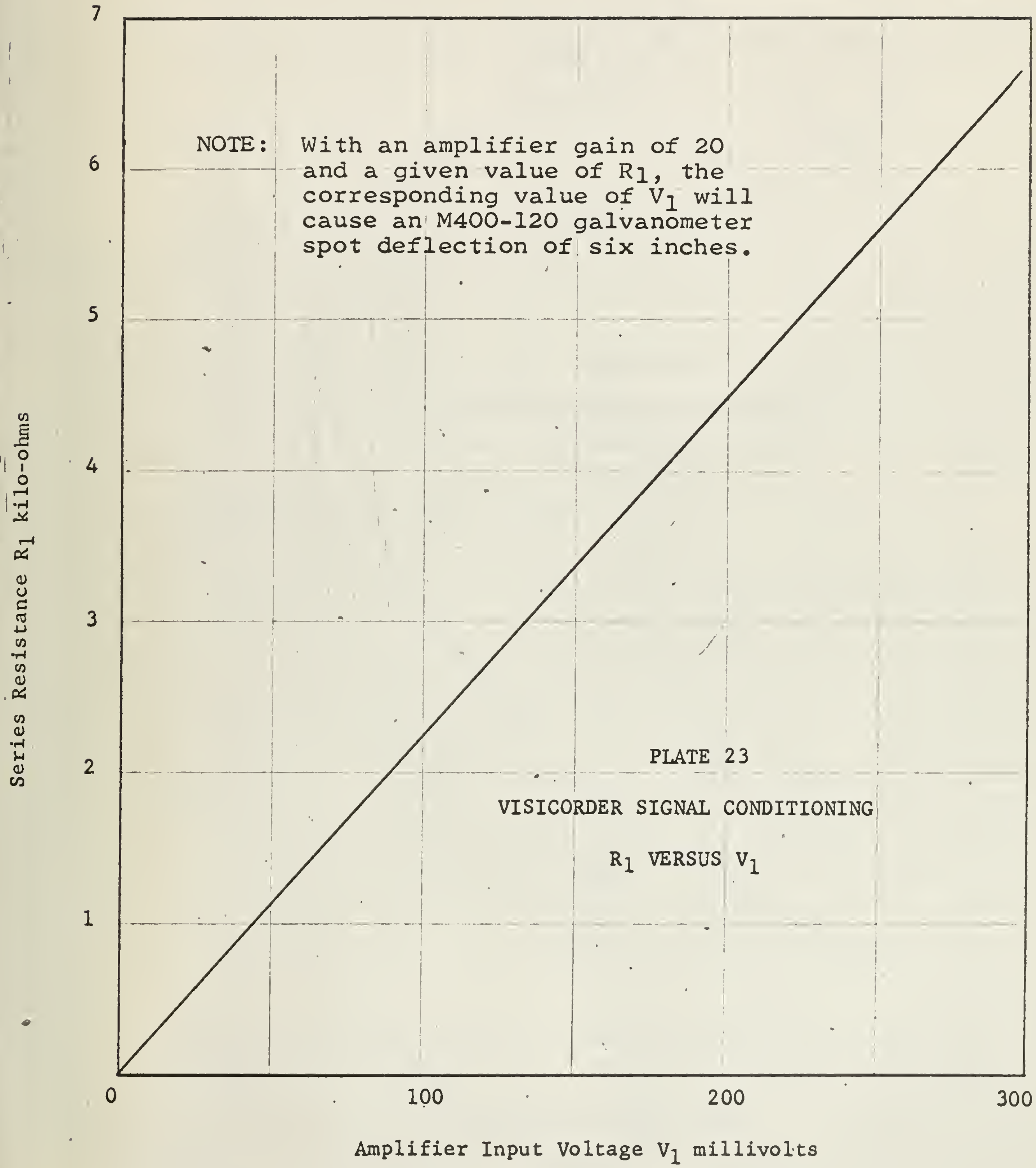
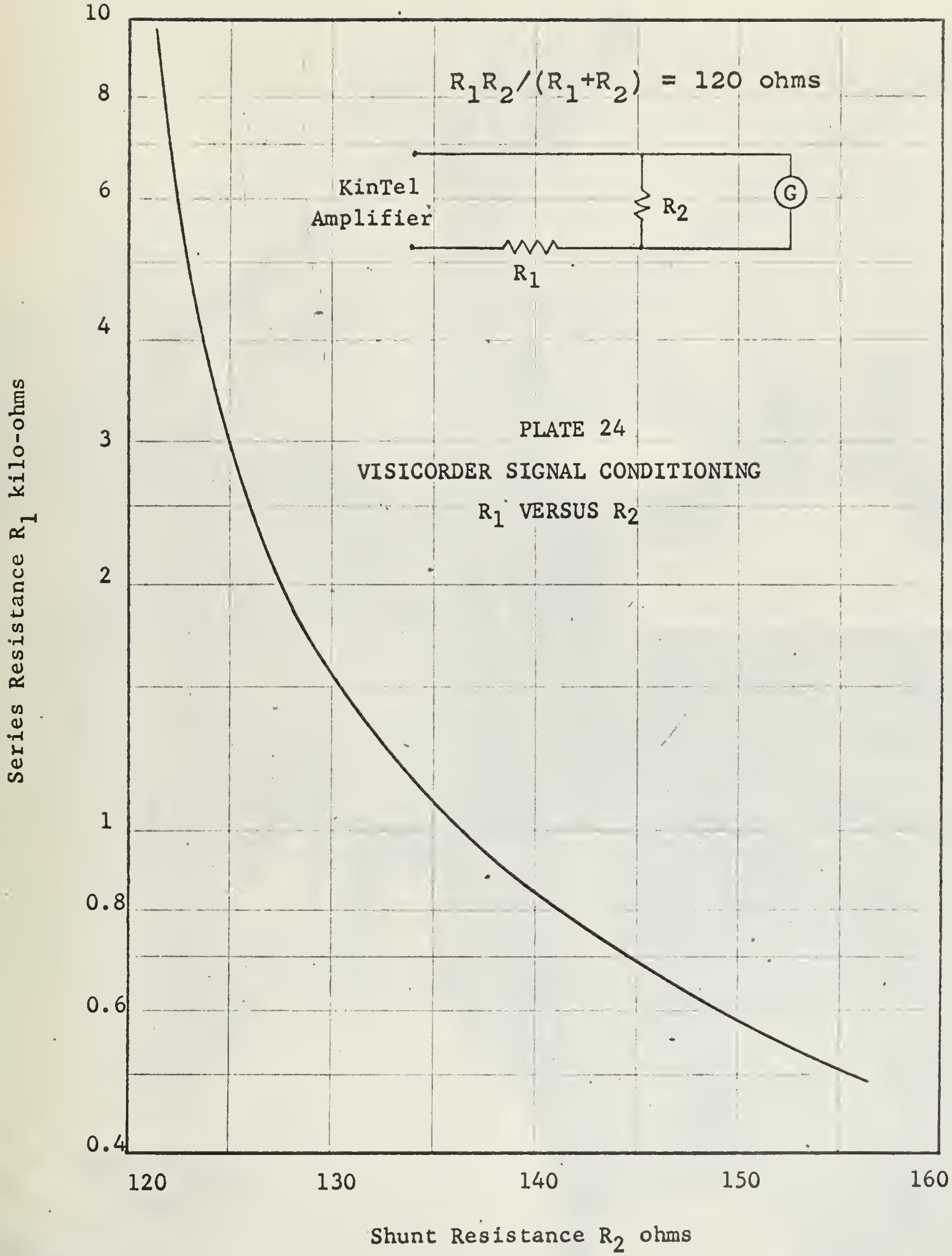


PLATE 23

VISICORDER SIGNAL CONDITIONING

R_1 VERSUS V_1



R1R2/(R1+R2) = 120 ohms
KinTel
Amplifier
R2
R1
G
PLATE 24
VISICORDER SIGNAL CONDITIONING
R1 VERSUS R2
Series Resistance R1 kilo-ohms
10
8
6
4
3
2
1
0.8
0.6
0.4
120
130
140
150
160
Shunt Resistance R2 ohms

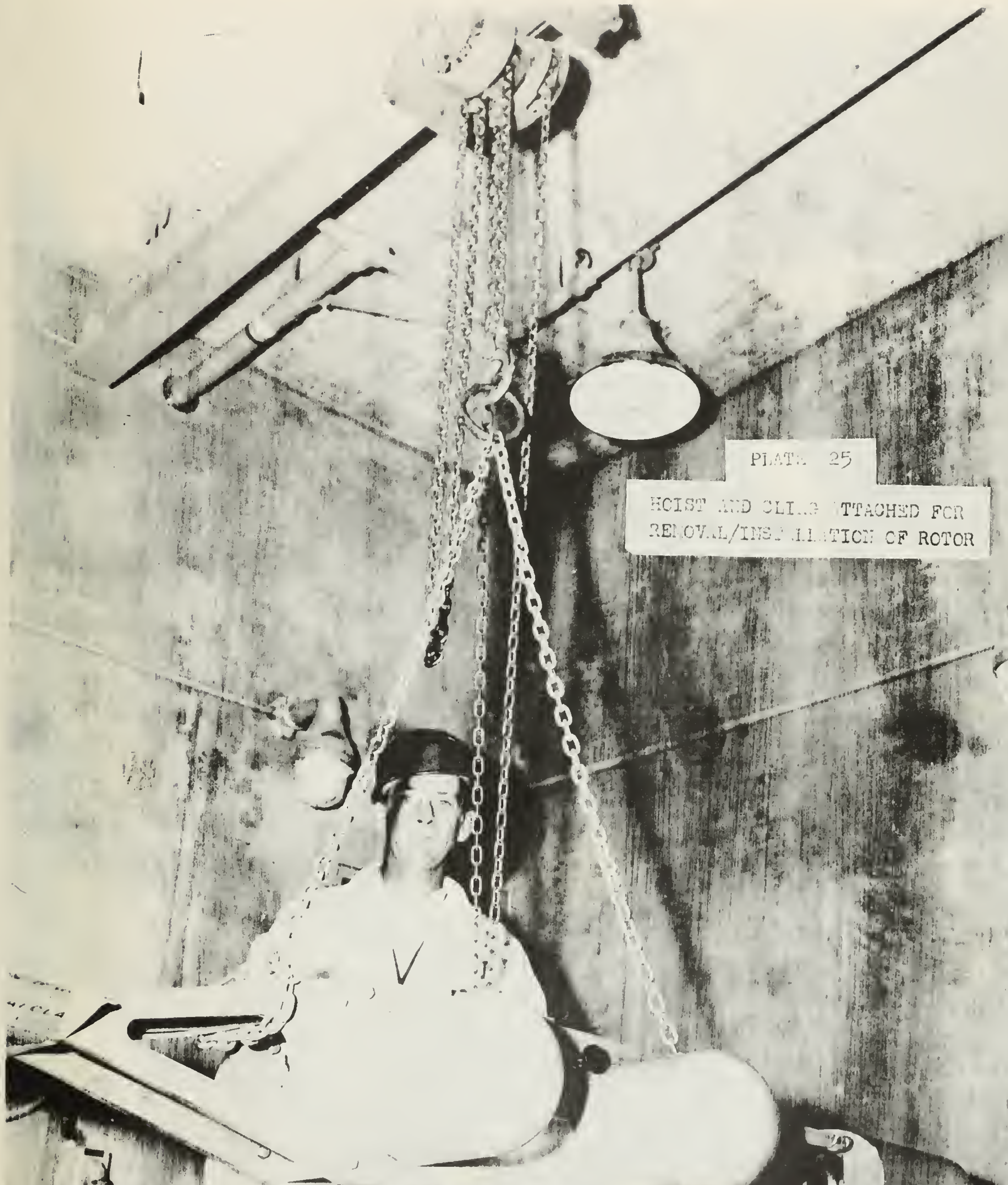


PLATE 25

HOIST AND SLING ATTACHED FOR REMOVAL/INSTALLATION OF ROTOR

PLATE 26

ROTOR BALANCING STAND